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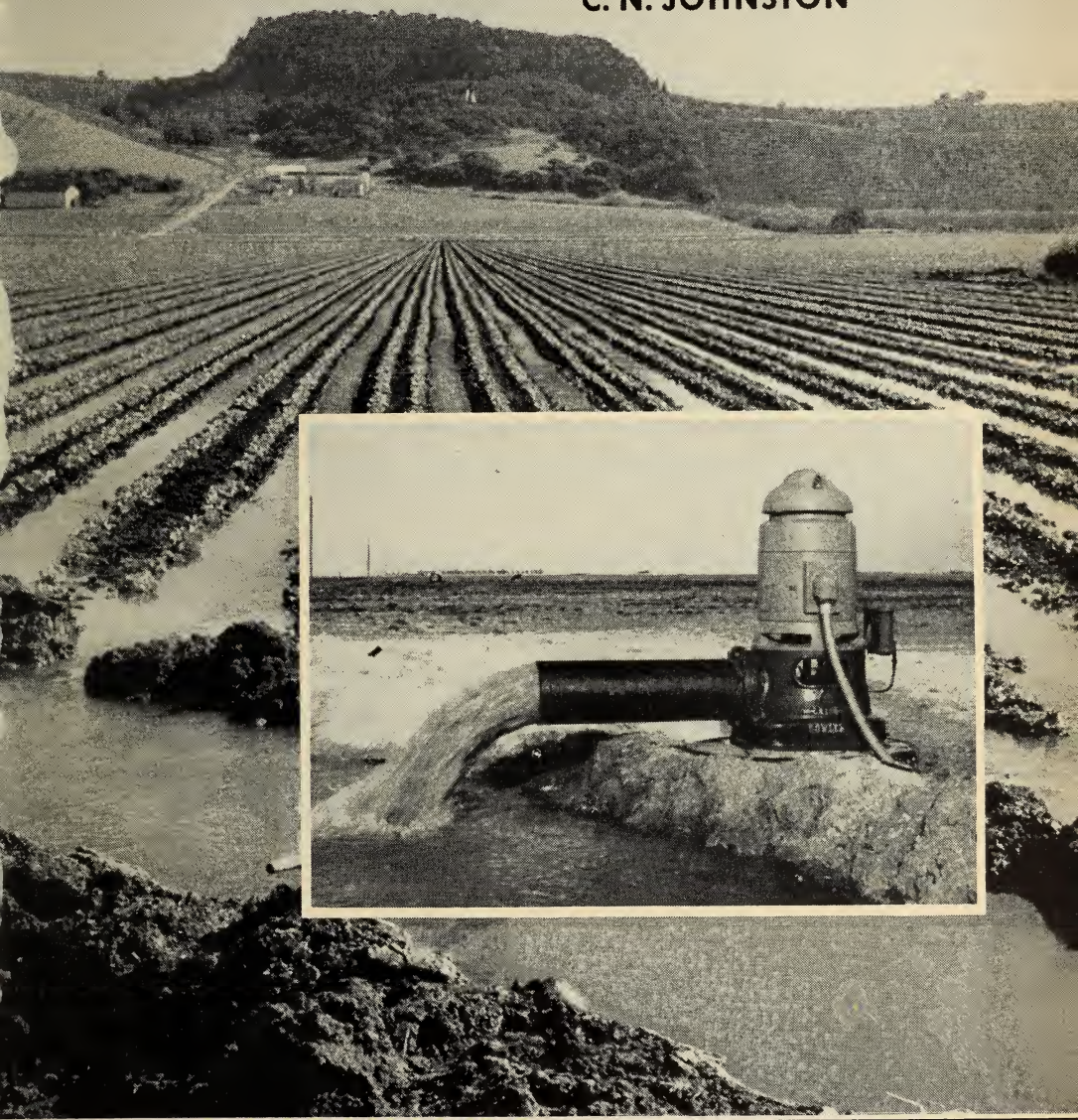
IRRIGATION PUMPS
Their Selection and Use

C. N. JOHNSTON



CALIFORNIA AGRICULTURAL
Experiment Station
Extension Service

CIRCULAR 415

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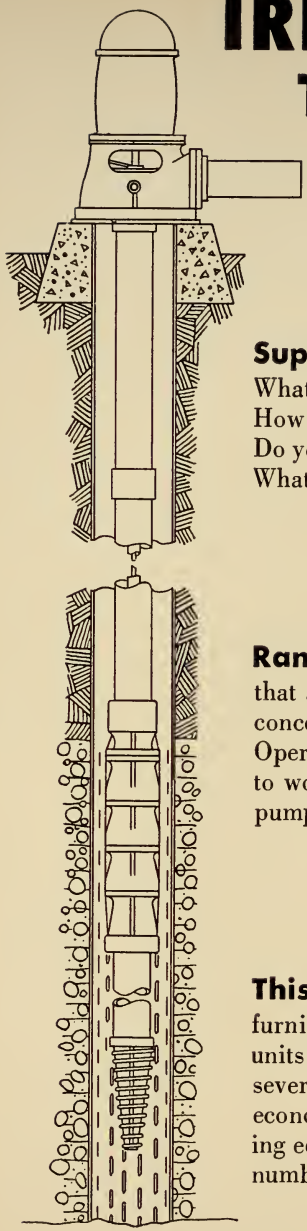

Suppose You Need to Irrigate

What kind of irrigation pump are you going to buy?
How will you fit your pump to your well?
Do you know how to work out power costs?
What do you need to know about care of pumping equipment?

Ranchers Who Use Irrigation Pumps Know

that although they have come into common use the problems concerning them have multiplied instead of becoming simpler. Operators need to understand this specialized equipment so as to work out for themselves the most efficient and economical pumping arrangements for their lands.

This Circular

furnishes as complete as possible a discussion of the pumping units themselves, the special attributes or characteristics of the several types of pumps, the application of pumps to wells, the economics of pumping, and the care and maintenance of pumping equipment. The seven equations used in the discussion are numbered where they appear and printed in heavy type.

The Author:

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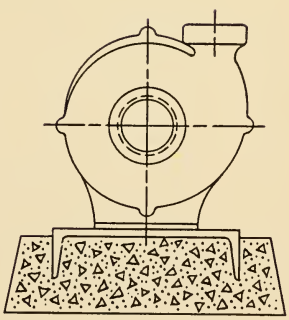


ALMOST ONE-HALF of the irrigated acreage in California is served by means of pumping. Much of the extension of irrigated land will be by this means. The selection of proper pumping equipment and its design are of great importance to the prospective buyer, who asks such questions as, "What type of pump should I buy?" and "Because I need a given amount of water, what size well and pump do I need?"

While variables not under complete control affect the answers to these questions, it is possible to describe these variables, show their part in the general problem, and provide a workable outline for anyone who may have doubts regarding his ability to make a wise decision in the matter.

The following pages describe the pumps themselves, then take up their performance characteristics, which are the clues to their adaptability to meet any set of conditions. The succeeding pages consider briefly the irrigation requirements of crops, the behavior of wells, and finally the application of pumps to wells, including typical pumping and power problems and their solutions.

SEVERAL KINDS OF PUMPS MAY BE USED

It is relatively safe to say that little irrigation water is being pumped by anything but high rotative-speed equipment of one sort or another. The few exceptions are mostly of limited capacity (20 to 50 gpm) in the form of windmills or power-driven reciprocating pumps, many of which are only variations of the wind-driven variety.

The high-speed rotative units are of two types: One type has a member, or impeller or runner, which is about the shape of a ship's propeller. This is a screw type. The other has a so-called centrifugal impeller. Its operation differs from that of the screw type in that the water is thrown out of it by the centrifugal force resulting from the high rotative speed of the drive shaft; hence the name.

Both types supply very large volumes of water for their size and weight as compared with the reciprocating pumps; they are therefore cheaper and more easily installed than reciprocating pumps of equal capacity. The centrifugal impeller is the more common of these two forms and will be considered first.

BY WAY OF EXPLANATION

Here are some abbreviations used frequently in this circular, together with the terms they indicate.

h—head
Q—flow
t—time
gpm—gallons per minute
cfs—cubic feet per second
rpm—revolutions per minute
h.p.—horsepower
w.h.p.—water horsepower
kw-h.—kilowatt hour

The centrifugal pump

Centrifugal force is a by-product of any rotation and is the tendency of all particles in the turning mass to continue in straight-line motion at any instant. The result of this force is noted when one has driven a car through mud and then picks up speed on the road. The mud on the wheels flies off all around the wheel.

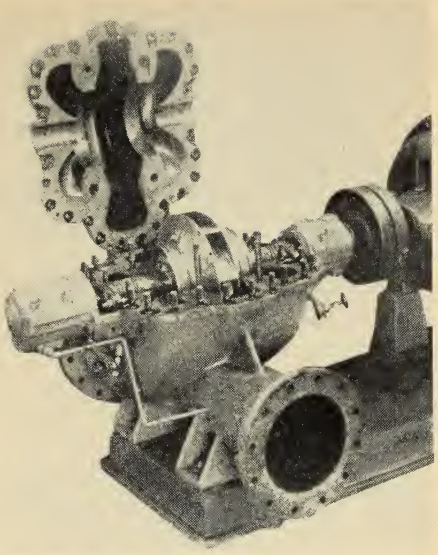
In a pump a shaft must supply the turning motion. On the shaft are mounted vanes which radiate outward. They generally curve as shown in the drawing. The vanes force the water to turn with the

shaft and thus generate centrifugal forces in the water. As a result the water is thrown outward from the eye or throat of the impeller to the outer edge of the vanes, and at the same time it is given a considerable amount of rotative speed about the center of the drive shaft. The combination of these motions given the water, which are measured as velocities, stores energy which we measure in one way as pressure or head. This is the ability to lift water to some given elevation.

Outside of the impeller there must be a pump case or housing to retain the water under control and to lead it into the discharge pipe, which carries it to the point of discharge. The drawing shows a pump case about the impeller. In shape it resembles a snail shell. The space around the impeller increases in cross section in the direction of rotation, so that in one revolution it grows from mere clearance to the full area of the discharge throat.

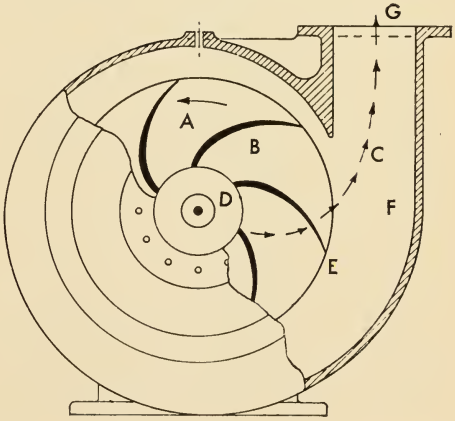
This type of case is called a volute because of its spiral shape, and it is designed to lead the water as efficiently as possible from the impeller to the discharge outlet. The drive shaft of a centrifugal pump can be mounted either vertically or horizontally. Horizontal mounting, as illustrated, is the more common arrangement.

At times it is desirable to obtain quite high pressures from a horizontal centrifugal pump. In this case two or more impellers may be mounted on the shaft. The discharge from the first is led to the throat of the succeeding and so on by appropriate arrangement of volutes and connecting castings in the case. Each impeller gives added energy to the water so that the final result is measured in pressure or head as many times greater than one impeller would give as there are impellers on the shaft.



Split-case double-suction centrifugal pump from discharge side, with top of case open to show parts.

Usually centrifugal pump impellers have, in addition to the vanes, two plates (one on either side of the vane edges)



Single-suction centrifugal with case or volute partially cut away. A—open space between vanes; B—vane; C—path of one particle of water stream passing through pump; D—inlet (or eye) suction port of impeller; E—outer edge of impeller; F—discharge end of volute passage; G—edge of discharge flange of pump.

Mixed-flow impeller of closed type.

cast integrally with the vanes. (A single side plate is occasionally used.) In this construction a little better control of the water is achieved than if the vanes alone are supplied. These plates (called shrouds) give additional stiffness to the vanes, which is a structural advantage.

Multiple-runner or multiple-stage pumps generally come with an even number of impellers on the shaft. Having an even number of runners is an advantage because it is possible to place pairs back-to-back. The advantage arises from the fact that the water in turning from the eye toward the periphery of the impeller creates a steady thrust on the plate at the back side of the vanes. With only one impeller (or an uneven number), this thrust must be resisted at a thrust bearing on the shaft. With back-to-back impellers little unbalanced thrust is left; cheaper thrust bearings are possible, and less bearing wear may be expected.

Unbalanced thrust in single impellers is counteracted, particularly where large capacity is to be provided, by casting a special pump case in which an impeller will operate having two opposing throats, or eyes. In this way the thrust from either side comes on the common center plate, and balanced operation is the result.

Such an impeller can provide double the flow that could be expected from a similar single impeller having but one throat and water passage of the same dimensions. The volute for a double-suction pump is exactly like that for a single-suction unit, except that it is larger so as to handle the increased capacity.

The turbine pump

A few of the horizontal centrifugal pumps and most of the vertical ones are called turbines because they have special cases that differ materially from the simple volute case. Turbines may have one or more stages and are more often multiple- than single-stage. The case contains stationary vanes that help direct the water away as efficiently as possible from the circumference of the runner to the discharge pipe. These vanes in the case of the pump mark it as a turbine. Deep-well turbines frequently have centrifugal pump-type runners.

The deep-well turbine case is more compact than the volute case, and that is why the deep-well turbine lends itself to use in the well casing, which is limited in diameter from both mechanical and economic considerations. In the deep-well turbine with two or more stages, the turbine vanes are necessary because without them the water leaving stage one would whirl up into stage two, turning with that impeller so that impeller two would be unable to act upon it at all. No work would be done, and no added pressure would be achieved above stage one. With the turbine vanes in place, the whirling water from the first stage is straightened out smoothly, flowing straight upward, before it enters the eye of the second stage, which can now set it spinning and eject it with added energy.

Screw-type pumps

Some single screw-type pumps are found in low-lift work delivering water from streams, ponds, or ditches with a lift of 2 to 10 or 12 feet. Well turbines rarely have screw-type impellers. These installations, as a rule, are multi-stage assemblies. Usually both shallow and screw-type installations have vertical shafts.

The screw impeller, like the boat propeller, is simply an inclined plane that when turned slides the water up over the plane and thereby lifts it. The inertia of the water, which tends to keep it in place, permits the plane to cut under it and lift it without whirling it too violently.

The low-lift screw is usually single-stage and is sometimes operated in a rather crude box-like (even wooden) housing. The factory product is generally better designed and better finished than these crude forms. The chief advantage of the screw impeller is its capacity to deliver more water than the centrifugal type of impeller for any given size. Performance characteristics of both types will be discussed later.

The screw deep-well turbine finds application to small-bore wells where a maximum amount of water is desired. The stationary turbine vanes are cast into the cases of these pumps as in the centrifugal type, and for the same reason.

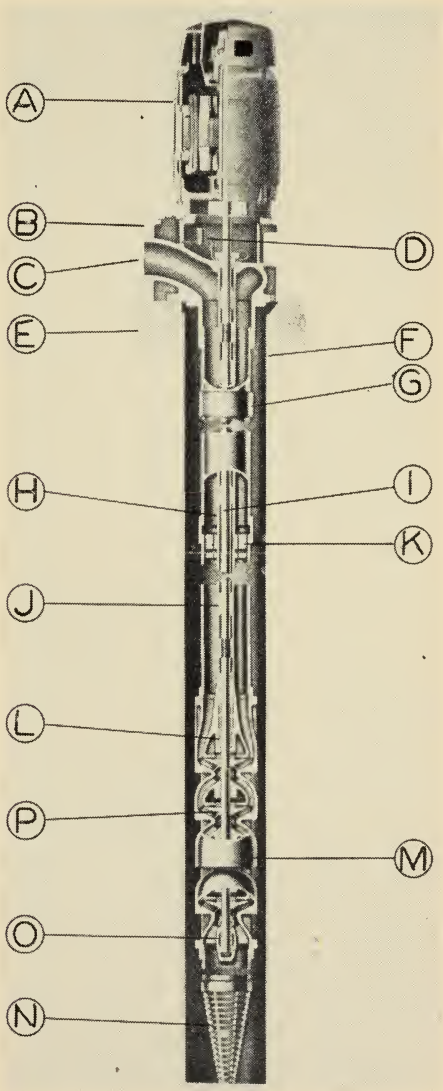
Combination or mixed-flow pump runners

If we took the common centrifugal pump runner and—supposing it were possible—bent it to give it a cone shape with the point of the cone cut off at the eye of the impeller and with the periphery of the impeller the big end of the cone, we would have produced a combination or mixed-flow impeller. Now if the vanes in the cone-shaped impeller are tilted slightly, they will slide under the water passing by, more like the flow through a true screw impeller.

These changes are accomplished in varying degree in the design of combination impellers to get some of the properties of both types for field application. They are used most frequently in deep-well turbines because they combine the feature of rather high discharge capacity, like the screw, with other features that are favorable to specific applications.

Mechanical features of various pumps

The centrifugal pump. Earlier illustrations have shown variations in horizontal centrifugal pump design. In the photo the volute case is bolted to the plate



Deep-well turbine assembly. A—motor; B—pump head; C—discharge; D—oiler; E—foundation base; F—well casing; G—column pipe; H—oil lubing; I—drive shaft; J—bearing; K—spider; L—upper bowl assembly bearing and seal; M—bowl; N—suction strainer; O—lower bowl assembly; P—runner or impeller.

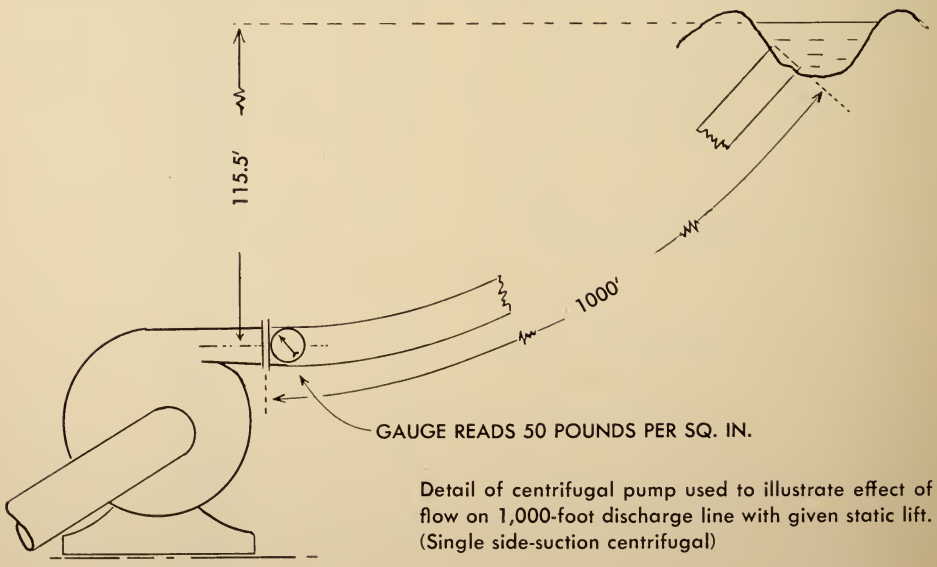
on the pump pedestal. As a rule the bolt holes are symmetrical so that it is possible to rotate the volute case about the pedestal plate, making the discharge point up, down, horizontal, or at almost any intermediate angle. The intake or suction opening stays centered on the shaft line, while the volute is swung on the bolt circle. This feature affords some flexibility to the pump when it is being installed. The drive shaft passes through the back of the pedestal assembly. It is sealed in a running fit in a stuffing box on the back side of the pedestal assembly so air cannot leak into the pump along the shaft.

One disadvantage, at times, in this form of pump (known as single-suction) is that the suction line and discharge line must be at right angles to each other. This may present a mechanical problem which must be anticipated before installation.

The volute case (shown on page 5) with the double-suction runner is much more complex than the single-suction design, for several reasons: (1) The case must be shaped to permit the passage of the suction stream around the discharge volute. (2) The case must be split horizontally to permit introduction of the drive shaft and double impeller. (3) The drive shaft must be supported in bearings at both ends, as well as the two stuffing boxes required to make a seal on the shaft when it enters and leaves the split case. While costly, these are not objections to this type of pump.

One advantage is that the suction and discharge ports of the pump are usually in line so that connecting piping may be run on a straight line from suction to discharge. This is often quite a desirable feature at installation time, and it generally affords a simpler-appearing piping installation than results when a single-suction horizontal centrifugal is used.

The true vertical centrifugal may be in fact a single-suction horizontal centrifugal case and runner set on end with the suction opening down, and having an extended shaft to permit power drive some distance above the pump proper. An elbow on the discharge of the volute directs the flow upward to the point of discharge. In this mounting the pump case is quite often submerged some distance below the water supply surface, and the inlet may or may not be equipped with a suction pipe.



Detail of centrifugal pump used to illustrate effect of flow on 1,000-foot discharge line with given static lift. (Single side-suction centrifugal)

The split-case double-suction centrifugal is occasionally mounted with a vertical-shaft drive similar to the single-suction installation, but it is not quite so adaptable to this service as the other type. Modifications to the suction port on the case or housing become necessary, and the unit is both bulky and heavy for some installations.

The deep-well turbine: Deep-well turbines as a group are similar in construction regardless of the type of runner. They are usually driven from the surface of the ground; hence the drive shaft has to be provided from that point to the bottom of the lowest stage, or bowl.

The water has to be conducted to the surface of the ground in a discharge pipe called the column. A 10- to 20-foot suction pipe with possibly a strainer at the end is usually provided. The turbine vanes in the case of each stage bring the water back to the drive shaft. Therefore the simplest location for the discharge column is immediately surrounding the drive shaft. Thus the drive shaft is in the center of the column pipe, surrounded by the discharged water. Under these conditions the shaft needs support every 6 to 20 feet, and these supports—bearings—need lubrication.

Rubber is adequately lubricated by water, and a stainless steel shaft works well against such bearings in this location. The rubber bearings are held in place by a thin ring threaded inside the column. The ring has 3 or 4 light arms running radially down the shaft to a housing which retains the rubber bearing itself. These assemblies are called *spiders*.

Rubber bearings are safely lubricated only if wet. If the pump is just being started, or has stood for a period unused, it is best to pour water down the shaft to wet the rubber bearings before starting. If this is not done injury will occur at the dry bearings before they are submerged in the discharged flow through the column pipe.

Before rubber bearings were available, a seamless steel tube was placed around the drive shaft in order to separate it completely from the water. This construction is still frequently used to avoid the hazard of injury to unlubricated bearings. Bronze bearings are furnished to center the shaft within this tubing, and a mechanically satisfactory installation results if oil bath or other positive lubrication is supplied.

Automatic oilers serve this oil requirement. You need only keep these oiler reservoirs filled regularly. The tubing is supported within the column pipe in light guides similar to the rubber bearing support (spiders) previously described.

The pump head and some form of power unit are located at the surface of the ground, which may be as much as several hundred feet above the pump bowls. The head is a heavy cast-iron assembly to which the column pipe can be suspended by screwing it in, and the water flowing from the column pipe can be turned and discharged horizontally. The assembly has a point of attachment for the oil tubing over the drive shaft (if oil tubing is present) and a stuffing box similar to the centrifugal pump to permit the drive shaft to pass through it either to the drive pulley or to a direct-connected motor.

There are no thrust bearings on a deep-well turbine at the bowl assembly; therefore to keep the drive shaft in tension and straight, and to keep the bottom of the runners from scraping on the bowls, a vertical thrust bearing is introduced at or near the top of the drive shaft but outside the pump head. For belt-drive pumps a combination thrust bearing is just below the pulley—as a rule on the top of the pump head. Some electric-motor drives have similar thrust bearings on the pump head, while others have the thrust bearing mounted on top of the motor. The motor itself in these installations has a hollow shaft. Further discussion will stress the importance of these thrust bearings. The photo shows a turbine assembly with most of these components.

OPERATING BEHAVIOR OF PUMPS

These terms are used in describing pump characteristics

Any given irrigation pump can be operated at varying speed, can lift water various heights, and can give a fixed rate of discharge for any given combination of the two variables—speed (revolutions per minute, or rpm) and lift (total head, or head). That is to say, if the pump is turning at 1,500 rpm and is lifting water 90 feet, it will give a certain flow. If, now, either the rpm or the lift is changed, a different flow will result. While these are the bare facts of the situation, before discussing the implications further we need a basis for understanding such terms as rpm, head, flow, power, and efficiency.

Rpm. The term revolutions per minute (rpm) needs no further discussion, but this factor affects some of the others as follows:

(1) The flow, or Q, varies as the rpm varies, or $Q \propto$ * rpm

(2) The head†, or h, varies as the square of the rpm, or $h \propto rpm^2$

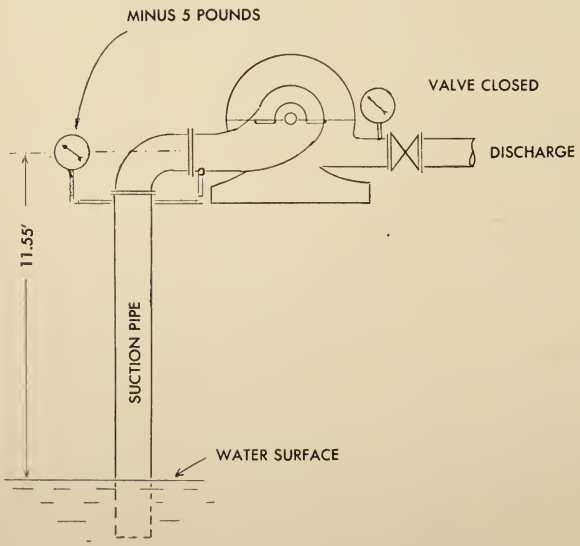
(3) The work done as water lifted varies as the cube of the rpm, or h.p. $\propto rpm^3$

The relationships above, called the affinity relations, are important to anyone having a belt-driven pump because the temptation is always present to change pulley sizes to change rpm (and thus the discharge from the pump) or to increase the lift it can operate against. Suppose the rpm before the change in pulley was 1,000 rpm and after the change became 1,100 rpm. Also suppose the h.p. before the change of pulleys was 20 and that the lift went up per the second above formula. What would be the theoretical h.p. at the new speed?

$$\frac{(1{,}000)^3}{(1{,}100)^3} = \frac{20}{\text{new h.p. requirement}} = \frac{(10)^3}{(11)^3} = \frac{10 \times 10 \times 10}{11 \times 11 \times 11} = \frac{1{,}000}{1{,}331}$$

* The symbol $\propto$ means "varies as."

† See further discussion below.



Illustrating measurement of suction lift. (Double-suction split-case centrifugal pump shown in detail)

Solution: New h.p. requirement = $(20 \times 1{,}331) \div 1{,}000 = 26.62$, which in this case is an increase of 6.62 h.p. resulting from 100 rpm added speed. In other words, a 10 per cent increase in the speed required 33.1 per cent increase in power under the conditions of this problem. Each set of conditions requires a complete computation, with the answers probably varying widely from those found in the above solution.

Head or lift. Lift has been defined as head, and it can be measured in units of pressure. It is therefore necessary to understand how to measure the lift accurately in order to calculate the pressure. For instance you can measure lift with a rule or tape of known length. Suppose the measured lift from the center of a centrifugal-pump discharge pipe to a point of delivery was found to be 115.5 feet, a vertical distance. Suppose the discharge pipe is 1,000 feet long and discharges into a ditch as shown in the drawing on page 8. If the ditch stays full, and if provision is made so that the water cannot escape from the pump, the discharge pipe will be full. If a pressure gauge is tapped into the discharge pipe at the pump and located on a level with the center line of the pipe at the pump, it will read 50 lb. per sq. in. with the pump stopped.

(This is the static pressure or lift.) In this case the following relations exist:

Head = 115.5 feet (as shown on tape) = 50 lb. per sq. in. (as shown on gauge).

(4) Therefore, 50 lb. per sq. in. = 115.5 feet head or 1 lb. per sq. in. = 2.31 feet head of water.

Now suppose the pump has been started. It is still possible to check the pressure-gauge reading and measure the net vertical lift from discharge-water surface down to the center line of the pump discharge. This taped distance is still 115.5 feet, but the pressure gauge will now read more than 50 lb. per sq. in. Assume the gauge pressure has become 62 lb. per sq. in. Then 12 lb. per sq. in. has been added to the static pressure, or lift, because of the friction of the water moving through the line. On the basis of equation 4 one would find 12 lb. per. sq. in. = 12×2.31 feet = 27.72 feet head which has been added to the static lift, making the total discharge lift 115.5 feet + 27.72 feet = 143.22 feet. The pressure gauge is therefore necessary to account for the total discharge lift or head, and the tapeline will not suffice.

You can, with proper training, compute the friction loss in a discharge-pipe line if the rate of flow and pipe size are known. Adding this friction loss to the measured static lift gives the total pumping head. This value should check very closely the value supplied under flow conditions by an accurate pressure gauge, but the method is not open to everyone, while an accurate pressure gauge can be used universally.

By the same reasoning a vacuum gauge which reads in negative pressure in pounds per square inch or in feet of head gives more accurate total suction-lift readings than are possible with a tape line. Negative pressure or negative head or suction lift can be described as follows: Suppose a centrifugal pump has been operating and the power is shut off at the same time the discharge valve is closed. The pump case and suction pipe are full of water at that instant, and if no air leaks into the case through the stuffing box the case will stay full.

The drawing on page 10 illustrates such a pump unit with dimensions given for further explanation. The distance from the water-supply surface to the center line of the vacuum gauge is 11.55 feet and, assuming the gauge is sensitive and accurate, it reads exactly 5 lb. per sq. in. vacuum or 11.55 feet vacuum, depending on how it is calibrated. The measured column of water (supply surface to center line of vacuum gauge) is hanging, in effect, from the level of the vacuum-gauge center to the supply surface and as a result creates negative pressure.

If the pump is started after this check the gauge will show an increase in vacuum because of friction losses in the suction pipe; and this new reading, in feet, is the total

suction lift to the center of the vacuum gauge. If, as illustrated, the vacuum-gauge center line is below the center of the pressure gauge, there is an increase of lift equal to the vertical distance between these two gauges, which must be added to the sum of the readings of the two gauges to get the gross total lift for the pump. For example, combining the data for static conditions from the two drawings above, we get:

Discharge pressure gauge		Suction vacuum gauge		Distance between gauges	Total static lift, feet
Lb./sq. in.	Ft.	Lb./sq. in.	Ft.	Assumed in this case*	
50	115.5	5	11.55	1.5	128.55

Sometimes the vacuum gauge is above the discharge-pressure gauge. In this case there is still a difference in elevation, or the vertical distance between them, but it is subtracted from the sum of the two gauge readings instead of being added. This is so because the vacuum reading when taken above the discharge-pressure reading overlaps some of the pressure recorded at the discharge gauge.

For example, suppose the vacuum gauge had been found to be 1.5 feet *above* the discharge gauge when checking the data from the two drawings, instead of *below* as found in the previous example. Then the total static lift would be 115.5 + 11.55 – 1.5 = 125.55 feet. Sometimes the vertical distance between vacuum and discharge gauges is considerable, and the correction for total pumping head is very important.

Flow. Flow or discharge refers to the amount of water being handled by the pump in a given unit of time. In irrigation the two terms, gallons per minute (gpm) and cubic feet per second (cfs), are most commonly used. A few other terms such as miner's inch, acre-foot per hour or day, and acre-inch per hour or day are also at times referred to. The following supplies the relationship of the terms:

1 gpm = 231 cubic inches per min. (cu. in./min.), or $\frac{231}{60}$ cu. in./sec. = 3.85 cu-in./sec.

1 cfs = 1,728 cubic inches per second

1 cfs is larger than 1 gpm in the ratio of $\frac{1,728}{3.85}$ = 448.83 or 1 cfs = 448.83 gpm

Since 448.83 is very close to 450, the expression is commonly written 1 cfs = 450 gpm. (It will be used in the following discussion.)

1 southern California miner's inch = $\frac{1}{50}$ of 1 cfs = just under 9 gpm

1 California statute miner's inch = $\frac{1}{40}$ of 1 cfs = a little over 11 gpm

1 acre = 43,560 sq. ft.; therefore 1 acre-foot = 43,560 cu. ft.

1 acre inch = $\frac{43,560 \text{ cu. ft.}}{12}$ = 3,630 cu. ft.

If 1 cfs flows steadily for 1 hour, then $1 \times 60 \times 60$ cu. ft. have been delivered, or 3,600 cu. ft. This value is so close to the 3,630 cu. ft. per acre-inch given above that one safely says 1 cfs for 1 hour = 1 acre-inch per hour, or 1 cfs for 24 hours = 24 acre-inches per day, or 2 acre-feet per day.

Power. All of the above terms involve volume and time. Volume fixes weight because a certain volume of any substance, including water, has a given weight. Power is the work done per unit of time and is expressed in terms of weight lifted in feet in a given time. (See box on page 13.)

* This value had to be assumed here but is carefully measured in the field, using a carpenter's level or the equivalent to transfer levels horizontally so that the vertical difference in elevation can be scaled.

1 h.p. = 550* foot-pounds per second = 33,000 foot-pounds per minute
1 h.p. = .746 kilowatt = 746 watts
1 h.p. hour = .746 kw-h. = 746 watts per hour

* Note: 550 foot-pounds per second might be 100 lb. lifted 5.5 feet in 1 second, or 55 pounds lifted 10 feet in 1 second, etc.

Returning to the expression at the start of this paragraph on power, you can understand that, in pumping, foot-pounds of work are accomplished in a given time; then power may be calculated if you know the weight of water per unit volume, as follows:

1 gallon = 8.33 pounds 1 cu. ft. = 62.4 pounds

From the above and the preceding relationships we derive water horsepower (w.h.p.) or the work done in unit time in delivering water by the following equations:

$$(5)\ \text{w.h.p.} = \frac{\text{cfs} \times 62.4 \times \text{total lift in feet (h)}}{550} = \frac{\text{cfs} \times \text{h}}{8.814}$$

$$(6)\ \text{w.h.p.} = \frac{\text{gpm} \times 8.33 \times \text{lift in feet (h)}}{33{,}000} = \frac{\text{gpm} \times \text{h}}{3{,}961}$$

Lift in feet in the above is the total pumping head, as measured. In this connection you should remember in the case of the deep-well turbine that the suction-pipe and column-pipe friction losses are generally unascertainable. Therefore it is customary to measure the distance from the pumping water surface in the well to the surface of the ground, and then add the total lift above the surface of the ground at the well to this figure, to obtain the apparent total pumping lift. This is a little unfair to the pump but the only practical way of doing it. The procedure will be discussed later in more detail.

A couple of examples of the use of equations 5 and 6 may make their application clearer. We can assume the following data have been determined from two pump tests:

Centrifugal pump test

Flow	Suction lift (h_s)	Discharge lift (h_d)	Distance between gauges	Total lift (h)
gpm	feet	feet	feet	feet
975	17.5	72.3	+ 2.2	92.0

Deep-well turbine test

Flow	Distance pumping water to ground	Total lift above surface of ground	Approximated total lift (h)
cfs	feet	feet†	feet
3.2	87.5	18.2	105.7

Computations of water horsepower

Centrifugal pump: w.h.p. (from equation 5) $= \frac{975 \times 8.33 \times 92}{33{,}000} = \frac{975 \times 92}{3{,}961} = 22.65$ w.h.p.

Turbine pump: w.h.p. (from equation 6) $= \frac{3.2 \times 62.4 \times 105.7}{550} = \frac{3.2 \times 105.7}{8.814} = 38.37$ w.h.p.

Any set of test data from a pump test can be used similarly to ascertain the water horsepower, which is sometimes called the output horsepower.

† In some cases total lift above the surface of the ground is measured with a pressure gauge, in which case the distance from the center of the gauge to the ground must be added to the gauge reading. At other times, with short discharge pipes, the lift is checked with a tape by measuring the vertical distance from ground line to water surface of the discharge.

There are two other terms that express power, namely brake horsepower and laboratory or test horsepower. These two terms are used by pump manufacturers to explain the behavior of their pumps. They are of interest to the buyer because they will be encountered in pump manufacturers' descriptive literature.

Brake horsepower is the power being delivered to the shaft of a pump at the pulley or flexible coupling. It is the actual effort being supplied by the motive unit (either electric motor or engine) at that point. It is useful to know this value or series of values for the performance of a pump, especially for a pump that is to be engine-driven, but it is not a measure of the amount of power the owner will purchase for an electric motor-driven unit, for example. The motor is not 100 per cent efficient (See below for a discussion of efficiency) and the input h.p. or kw-h. per hour to the motor is what affects the true cost of power to the owner.

It is safer to convert factory data showing brake h.p. to input h.p. by correcting the power as follows:

$$\frac{\text{brake h.p.}}{\text{per cent motor efficiency}} \times 100 = \text{input h.p.};$$ and input h.p. $\times .746 =$ kw-h. per hour input. Later in the discussion, curves showing pump-performance characteristics are supplied with the input h.p. and input kw-h. per hour shown, but in some instances not the brake h.p. In these cases, the brake h.p. would be input h.p. $\times \frac{\text{motor efficiency.}}{100}$

Laboratory or test horsepower is associated principally with turbine pumps. It is actually the brake h.p. required at the factory for a test of the bowl assembly alone. It does not include losses resulting from the long drive shafts to be found when the bowls are put in deep wells. Hence the figures are low and not useful for checking true input horsepowers.

Efficiency. Everyone is familiar with the term efficiency as the ratio of recovery to some original effort; in pumping it has a similar meaning. The water horsepower divided by the input horsepower gives the decimal value of this ratio, and multiplying this decimal by 100 gives the per cent efficiency. The equation can be written thus:

$$(7)\ \frac{\text{output h.p.}}{\text{input h.p.}} \times 100 = \text{efficiency}$$

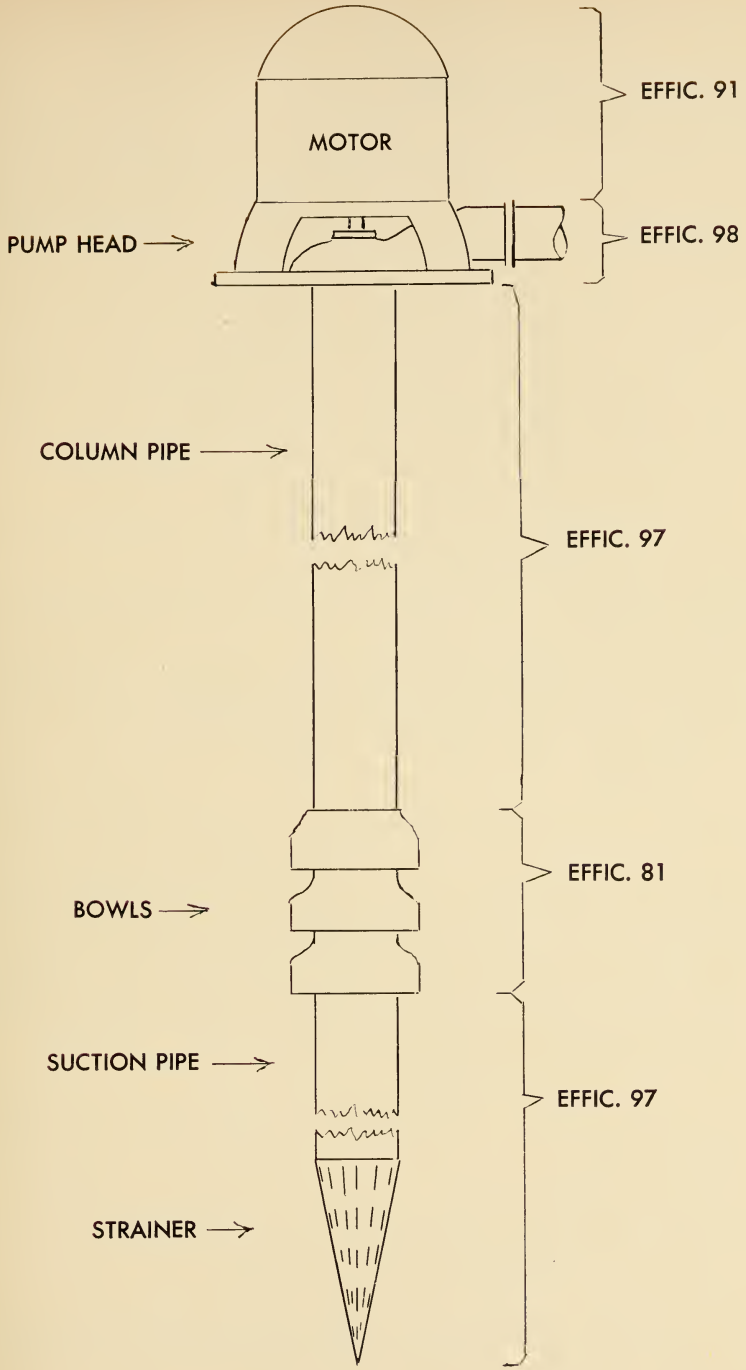
Suppose in the case of the turbine pump for which an output of 38.37 water horsepower was computed in the preceding discussion that the input horsepower was 62.0. Then the efficiency is $\frac{\text{output, } 38.37}{\text{input, } 62.0} \times 100 = .6189 \times 100 = 61.89$ or 61.9 per cent.

The term over-all efficiency is common in pumping work and particularly for electric motor-drive units. It includes the losses in efficiency in the motor as well as all the losses in the pump unit itself. The owner pays for power on the basis of over-all efficiency; hence the term is important in computing or estimating pumping costs.

We can illustrate the derivation of over-all efficiency by a drawing in which the individual efficiencies of the various parts of the unit are noted. Here, starting with the suction and strainer with an efficiency of 97 per cent, the bowl assembly with an efficiency of 81 per cent, the column pipe with 97 per cent, the head of the pump with 98 per cent, and the motor with 91 per cent efficiency, one computes an over-all efficiency of 67.9 per cent by multiplying all of the efficiencies of the component parts together, or $(.97 \times .81 \times .97 \times .98 \times .91) \times 100 = 67.9$ per cent.

When asking for bids on new pumping equipment, specifiy that over-all efficiencies be supplied whenever possible.

(This is suggested because laboratory or test efficiencies are likely to be supplied



This figure shows roughly how losses occur in such a deep-well pump. For example of the full power supplied the motor, 91 per cent becomes useful work. Of the energy reaching the pump head only 98 per cent is useful, and so on down to the suction and strainer assembly.

otherwise, and confusion and possibly error may result from attempting to convert these data into equivalent over-all efficiency and further into input horsepower.)

Examples of the interrelationship of the pump characteristics

Often you may have only a portion of the data from a pump test, but because the characteristics are all interrelated it is possible to derive the missing values. Three examples follow to illustrate what is meant and to show how partial data can be used to derive missing portions to complete the picture for any given pump.

Example 1: Assume we know the discharge (Q) = 1.7 cfs; the total head pumped against, including both suction and discharge lifts, is 97 feet; and the motor is consuming 34 horsepower at the meter. It might be desirable to know the over-all efficiency, the water horsepower, and perhaps the net horsepower delivered by the motor to the pump so that a different type of drive could be substituted if necessary.

Water horsepower is pounds lifted in feet in a fixed time, and using equation 5 we find the following by substituting the numbers in this problem:

$$\frac{\text{cfs} \times 62.4 \times \text{total lift in feet}}{550} = \frac{1.7 \times 62.4 \times 97}{550} = 18.7 \text{ w.h.p.}$$

From equation 7 by the same process we get

$$\frac{\text{output horsepower}}{\text{input horsepower}} \times 100 = \frac{18.7}{34} \times 100 = 55 \text{ per cent over-all efficiency.}$$

Motor manufacturers supply efficiency curves for their products, and pump vendors as well as power companies have these data; therefore the efficiency of the motor can be approximated at this load of 34-horsepower input. Suppose it is found to be 90 per cent. Then $\frac{\text{output h.p.}}{\text{input h.p.}} \times 100 = 90$ per cent $= \frac{\text{output}}{34} \times 100$. Therefore output h.p. $= \frac{90}{100} \times 34 = 30.6$ h.p. This is the so-called brake horsepower required by the pump when lifting 1.7 cfs, a total height of 97 feet. We would have to supply a power unit capable of delivering this much power continuously to the drive shaft of the pump.

Example 2:* Suppose we wanted to determine how many gallons per minute a pump would supply if the total pumping head were 105 feet, the over-all efficiency of the unit were 55 per cent, and the motor could safely use a maximum of 47 horsepower at the meter. It is possible to ascertain the gpm (Q), the actual water horsepower (w.h.p.), and the kw-h. per hour drawn by the motor.

Using equation 7 again, $\frac{\text{output h.p. (w.h.p.)}}{\text{input h.p.}} \times 100 = 55$ per cent $= \frac{\text{w.h.p.}}{47} \times 100$

$$\text{w.h.p.} = \frac{55 \times 47}{100} = 25.85$$

By using equation 6 this time (because gpm are involved) we have:

$$\frac{\text{gpm} \times \text{lift}}{3{,}961} = 25.85 = \frac{\text{gpm} \times 105}{3{,}961}$$

* Note: This example like several others here is set up to illustrate the method of obtaining certain operational data by computation, when other information is available as a starting point. Here the over-all efficiency, the lift, and the actual input horsepower to the meter are known from the following sources: the over-all efficiency presumably from some pump manufacturer, the lift from known field data, and the motor-input horsepower from previous experience, or from the manufacturer or his representatives. We may often wish to learn with these data available how much water can be handled, and this problem gives the procedure. Variations in other problems meet different situations encountered in the field.

$$\text{gpm} = \frac{25.85 \times 3{,}961}{105} = 975$$

The kw-h. per hour used by the motor is $47 \times .746 = 35.06$ kw-h. every hour.

Example 3: Against what total head can a pump unit with an over-all efficiency of 58 per cent lift 2.2 cfs if the safe motor input is 28 h.p., and what is the water horsepower?

Using equation 7, as in example 2,

$$\frac{\text{output h.p. (w.h.p.)}}{\text{input h.p.}} \times 100 = 58 = \text{w.h.p} \times 100$$

$$\text{w.h.p.} = \frac{58 \times 28}{100} = 16.24 \text{ water horsepower}$$

Substituting in equation 5, we get

$$\frac{\text{cfs} \times 62.4 \times \text{total lift}}{550} = 16.24 = \frac{2.2 \times 62.4 \times \text{total lift}}{550}$$

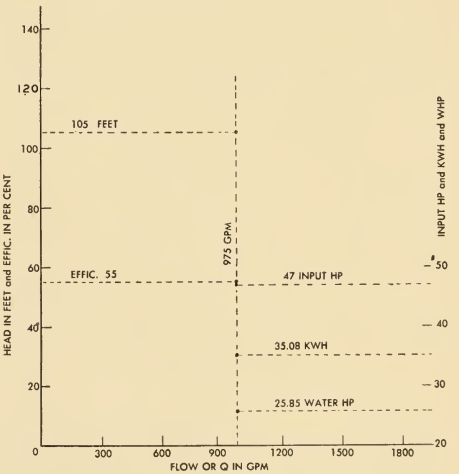
$$\text{Total lift} = \frac{16.24 \times 550}{2.2 \times 62.4} = 65.06 \text{ feet}$$

It can be seen that there is a certain order to the solution of these problems, because the calculations are not taken in order of the listing of the unknowns in the examples. We have to apply the equation that fits the conditions and have only one unknown left to get a solution. It is a fact, too, that these computations have been possible because all of the terms like flow (Q), lift (h), efficiency, etc., are interrelated in a given pumping unit, so that if part of them are available, as a rule, the rest can be calculated. We can do little if both head and Q are missing from the data, but, with only one lacking, many other values may be calculated, as well as the missing value of h or Q.

Pump performance or characteristic curves

How these curves are drawn. The previous discussion has considered a given flow or Q from a pump; and the several related factors such as head, efficiency, water horsepower, input horsepower, and over-all efficiency have been provided originally or found by computation.

Taking example 2 above as typical, we might decide to make a picture of the data, both supplied and computed on a graph. Custom has put the flow, or Q, on the bottom of the graph and the scales for the other factors up the left or right side of the graph (at right). This figure is finished so far as the data from example 2 are concerned. The method for completing it in graphical form is as follows and based on these data: Q = 975 gpm; over-all efficiency = 55 per cent (minimum); kw-h. used per hour = 35.08; h = 105 feet; input h.p. = 47; water h.p. = 25.85.



Graph of data from one test of a typical pump. Note there is one horizontal scale (in this case, gpm). Also note that there are actually 5 vertical scales, but that in this drawing head and efficiency (over-all) are combined on the left side and that the three power scales are combined on the right. Any other combination might have been worked out for the vertical scales, but Q would stay at the bottom or be plotted as the only horizontal factor in the graph.

A suitable scale in gpm is selected for the bottom line, placing 975 gpm in this case about the center of that line. Then to either the left or right sides of the chart another group of scales is planned to take care of the rest of the factors. One scale generally takes care of input, output horsepower, or w.h.p., and kw-h. demand. Sometimes efficiency also uses this scale. In the graph shown, however, efficiency and head are on the same scale. This was purely arbitrary and subject to the draftman's decision in order to separate the points reasonably well. A light dotted line is run straight up from the 975-gpm point on the bottom line. Then starting at the head-and-efficiency scale on the left, horizontal lines are run to the right from 105 feet head and 55 per cent efficiency points on this scale to intersections with the dotted line representing 975 gpm.

We have now located two points that represent respectively the h – Q and the efficiency – Q characteristics of the pump for 975 gpm. By the same process except from right to left the two horsepower points and the kw-h. point with respect to Q are spotted on the graph. The graph is now done. We have to assume that this flow and the related points plotted are the result of some constant rpm so that no variables arise from changing rpm. (See page 10, equations 1, 2, and 3, which indicate effect of varying rpm.)

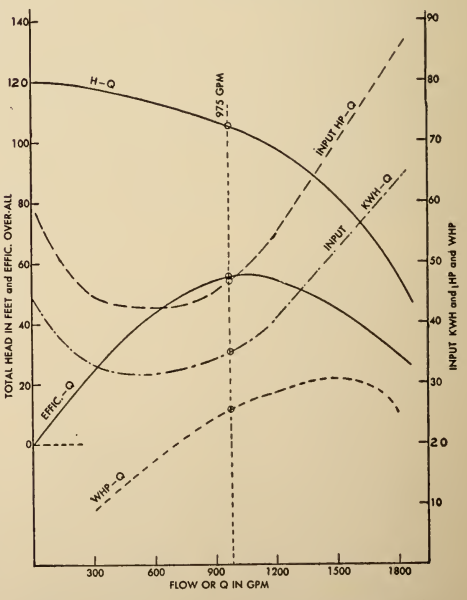
If the efficiency of the motor were known, it would be possible to derive the brake horsepower also as follows:

$$\text{input h.p.} \times \frac{\text{motor efficiency}}{100} = \text{brake h.p.}$$

This term could be plotted against Q along with the other power units already shown. Brake horsepower would be the work done by the motor at the pump shaft.

Suppose more data were obtained from the pump while it was turning at constant speed and while a discharge valve was opened and closed so that the total lift varied considerably. Then many more test points would be available and other points could be computed. As the discharge valve was being closed the total lift would tend to increase and the flow rate would diminish. Correspondingly, other points would change up or down in value, depending on the type of pump under test. The value of these data lies in the fact they may also be plotted as in the graph on page 17 as a series of points for h – Q, efficiency – Q, etc., that will tend to form smoothly curved lines representing the characteristic of the pump over its full range at that speed.

Pump characteristic curves in general: This graph shows a group of pump characteristic curves. Note that curves pass through the 975-gpm line at the same points as those on the first graph. This was done on purpose to illustrate how those test data might have been completed. These curves may duplicate the performance of some model centrifugal or deep-well turbine pump at some fixed rpm, but they cannot be applied to any specific pump without testing and checking it. In fact, each pump has its own characteristics, and its curves cannot be used to study any other unit.



Finished graph of characteristic curves resulting from series of tests on same pump for which one test was graphed on fig. 10.

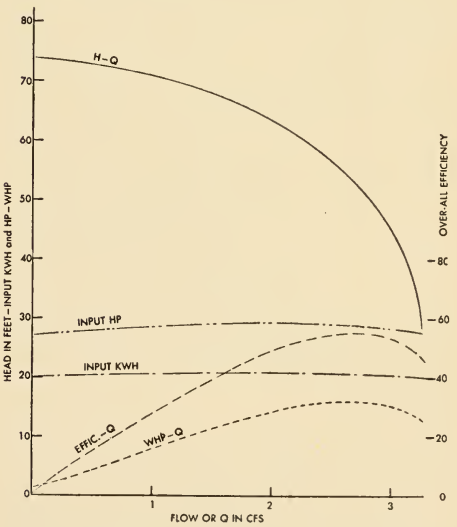
The shapes of these curves are of interest because, if the pump is discharging any given quantity of water within the Q scale, a line drawn straight up from that point on the Q line intersects all the curves, and the points of intersection locate the values of h, efficiency, etc., at which the pump must have been operating. Thus if the flow becomes quite small, say 100 gpm, the head must have become about a maximum, the efficiency has approached zero, and the other factors have also changed from the 975 gpm condition.

Another detail of importance is the fact that when zero Q is being pumped, or when the discharge valve is shut off tight (or if the total lift exceeds 120 feet for this pump), the pump supplies just 120 feet head and no more. Shutting off the discharge valve can do no injury to the pump itself for that reason. Each pump will have its own shut-off head, which it will never exceed so long as the rpm stays constant. Changing the rpm shifts all the curves somewhat and establishes a new shut-off head. Direct motor drive pumps maintain constant rpm conditions.

Over to the right of the chart, above 1,800 gpm flow, the head that the pump can develop has fallen to 54 feet. This means that if the total lift is actually 54 feet the pump will be delivering 1,800 gpm. The three power curves, and particularly kw-h. per hour input and horsepower input, are not good at all in this pump for the reason that the minimum horsepower input corresponds to 600 gpm, while at zero and at 1,500 to 1,800 gpm output the horsepower input has risen to double or more that required at 600 gpm. If the pumping head varied widely for this installation, one would have to supply over 80 horsepower capacity to move 1,800 gpm, 60 horsepower at close to shut-off capacity, and only about 30 when moving 600 gpm. If an 80-horsepower motor had to be put on the pump costly operations would be induced as well as high initial costs.

One conclusion from the above discussion and from inspection of the graph is that this pump running at constant speed has fixed operating characteristics—so that if the flow is, say, 1,200 gpm, the total lift must be 97.6 feet and no other, and that corespondingly water horsepower, input horsepower, kw-h. per hour, and over-all efficiency must be fixed as indicated by the individual curves where they cross the 1,200 gpm line. There is no escaping this fact; at any given rpm the total lift by the pump controls the discharge (Q), and all the rest of the factors are then set.

Characteristic curves for centrifugal-type runners: The graph shown below is a set of pump characteristic curves rather typical of the cenrtifugal-type runners in either horizontal or deep-well turbine cases. A notable difference appears between the input and kw-h. per hour power curves on this graph and those on the previous graph. Here these two power curves are quite flat. This form is very desirable for the reason that it is impossible to overload the motor under any conditions, if the motor is sufficiently large to handle normal pumping loads out near the peak efficiency.



Characteristic curves from a pump having centrifugal-type runners.

Present h (2 stages), feet	Q or cfs		Head per stage h, feet
	Present	Per stage	
74	0	0	37
71	1.0	1.0	35.5
63.4	2.0	2.0	31.7
45	3.0	3.0	22.5

If we wished to assume that this graph came from a deep-well turbine having a centrifugal runner, it is probable there were two runners or stages in series producing the h – Q curve. This statement is predicated on the actual situation in the field for nominal sizes of deep-well pumps, where the head per stage will be somewhere between 20 and 40 feet. In the case of this pump, and assuming it is two stages, then each impeller or stage is delivering half the head shown on the h – Q curve on the basis of the table above.

Should a third stage be supplied with this pump, the total head would increase by 3 times the corresponding head per stage for each discharge rate, or if plotted on this graph the present h – Q curve would be shifted upward, at each Q, the indicated head per stage given in the table. The power curves would be shifted correspondingly, but the over-all efficiency curve would shift only slightly.

This can be put another way as follows: Suppose we had available a single, a 2-, and a 3-stage pump of this design, the test data shown below would result if we decided on, say, 3 cfs as the required constant discharge:

For these pumps it is logical from equation 5 that the power values should change as a multiple of the number of runners if the head changes in that ratio, because in the equation the only thing that will change will be head. This can be illustrated as shown at top of page 21.

The significance of these computations and the preceding table lies in these facts: (1) To meet increasing total pumping lifts additional stages can be added to a given pump. (2) Such an addition to the pump will in all probability increase the input horsepower (this will be certain if the delivery rate is kept the same as it was before after adding the new runner or runners). (3) This added horsepower

Pump Characteristics of One-, Two-, and Three-stage Identical Runners at 3 cfs Discharge

Characteristic	Single stage	Two stage	Three stage
Total pumping head	22.5 feet	45 feet	67.5 feet
Over-all efficiency (per cent)	54	54	54*
Water h.p.	7.75	15.5	23.25
Input h.p.	14.2	28.4	42.6
kw-h. input per hour	10.25	20.5	30.75
Probable size motor (normal h.p.)	15	30	40†

* Note: Sometimes there is a slight improvement in over-all efficiency as the number of stages increases.
† Note: This mathematical discrepancy (45 h.p. seems to be indicated) will be explained later in the discussion.

$$\frac{\text{cfs} \times 62.4 \times \text{total lift in feet}}{550} \qquad \text{equation 5}$$

Substituting the cfs and total lift from the preceding table,

$\frac{3 \times 62.4 \times 22.5}{550} = 7.66$ w.h.p. — One runner or stage

$\frac{3 \times 62.4 \times 45}{550} = 15.3$ w.h.p. — Two runners or stages

$\frac{3 \times 62.4 \times 67.5}{550} = 22.97$ w.h.p. — Three runners or stages

input will probably require a new motor for driving the unit (this is often the case), and possibly a new and heavier drive shaft will be needed.

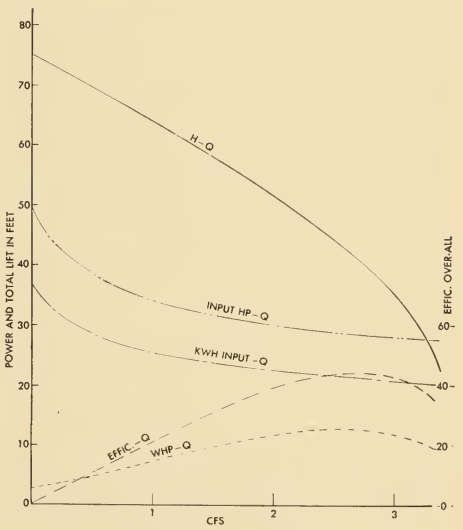
Unfortunately more runners cannot be added to a pump, short of factory installation. A complicated machining, balancing, and precision-fitting job needs to be done in assembling a multistage pump of any type, and adding new runners requires factory facilities. The deep-well-turbine type is the only one actually suitable for such changes because of the simpler case design in that unit.

Characteristic curves for screw-type impellers: As stated earlier the screw impeller develops a relatively low lift per stage. The graph (at right) gives the characteristic curves for a typical multi-stage screw pump. As in the case of the centrifugal-type pump, the h – Q curve for the screw pump is the sum of the heads produced by the individual impellers. Hence it is possible to analyze the curves for the screw pump in the same manner as the centrifugal was handled in Section C above.

Briefly stated, on the assumption that there are, say, 5 impellers in this screw pump, the efficiency curve per impeller would be almost, if not, identical with that on the graph, while the h – Q and the three power curves would be one-fifth as high on the vertical scale as they are at present for the assumed 5-stage pump.

It has been noted above that the shape of the power-input curves is of special interest. These two curves (input h.p. and kw-h. per hour input) on the graph at right are typical of screw-type impeller pumps, which almost all show the tendency to demand more horsepower at or near zero flow than when pumping a considerably larger flow out in the neighborhood of the peak or high point of the efficiency curve.

This characteristic has some disadvantages for the owner, which can be illustrated by setting up an imaginary field situation. Suppose this pump were operating satisfactorily at a discharge rate of 3 cfs. The input h.p. would be about 28, and a 30-h.p. motor would be doing the job without any trouble. It is now decided to plant a lot of small trees and carry water to them with a tank truck, which will involve the use of about ⅕ cfs (90 gpm) for filling the tank at a reasonable rate. To deliver just .2 cfs the pump will have to be discharging against about 73-foot head,



Characteristic curves for a pump with screw-type impellers.

and the input h.p. will become 43, the over-all efficiency about 4 per cent.

The 30-h.p. motor will burn up quickly under this load, and so the decision to use the pump for this tank-filling job needs some practical reconsideration. Following the same reasoning, the owner of this pump always faces the possibility that someone will partially or completely shut off the discharge valve—by mistake or otherwise. If this ever happens while the pumping unit is in operation, a burned-out motor will result.

The immediately preceding is an indictment of the screw-type pump only for the special condition listed (too high head). As has previously been noted, it produces a maximum flow with a minimum of pump-case diameter. When the pumping lift is steady and canot be interfered with and maximum production is required, it is valuable. The operator who has such a pump need only remember this high head-low flow hazard to avoid trouble. The buyer of new equipment that must meet a wide range of flow rates should avoid such a pump. The characteristic curves which can be obtained for any pump being considered tell the story.

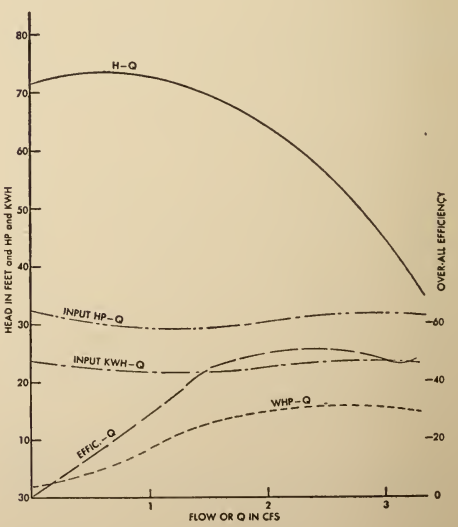
Another difference between the screw characteristic and the centrifugal pump characteristic lies in the h–Q curve shape. Comparing the two h–Q curves on the graphs on pages 19 and 21, we see that the screw type tends to be steeper. This means that for the screw pump a given change in pumping head will cause less change in discharge values than a corresponding change in head for a centrifugal pump—another argument in favor of the screw pump.

When for one reason or another the over-all pumping lift changes quite a bit during the season, the screw-type impeller provides a more nearly constant flow than the centrifugal type. This is often a very good argument for the use of the screw-type impellers in wells, if the shut-off or near shut-off hazard can be eliminated.

The curves on these last three graphs, which will be discussed later, can be considered as coming from rather old units in which the impellers have worn, considerably reducing the over-all efficiencies. Either type can be bought now to provide 60 per cent or better over-all efficiency, but in time these efficiencies tend to become lower.

Characteristic curves of mixed-flow impellers: Mixed-flow impellers have been described earlier as a combination structurally between screw and centrifugal types. Their behavior as indicated by the characteristic curves is also a combination of the two, with peculiar traits sometimes added. The graph below shows the characteristic curves of a combination or mixed-flow runner pump having h – Q characteristics at zero flow somewhat like those given in the two preceding graphs, which might have been the "parents" of this mixture. That is, the pump might be a combination of the other two pumps. As we noted earlier, this is presumably an old pump that has worn, and so the efficiency is considerably lower than should be expected of a new unit.

Several things might appear to come logically from mixing the characteristics



Mixed-flow impeller pump characteristic curves.

of two such different runners: (1) The head developed per stage for the screw might be increased. (2) The power curves might not swing up near zero flow as excessively as they do in the graph on page 21. (3) The h – Q curve might become a combination of the h – Q curves for the two "parents," or at least different from either. All these things are true, and in some cases the argument for using the mixed-flow runners hinges on the benefits derived as a result of changed characteristics that create most effective performance patterns. Sometimes the combination design produces somewhat objectionable performance features.

The graph at left can be used to discuss both desirable and objectionable features. Of the desirable features the relatively flat input-power curves are of course noteworthy. The h – Q curve is quite similar to that for the centrifugal runner pump, except for the fact that the maximum head (highest point on the curve) is some distance out to the right of zero flow. This feature is quite often found in mixed-flow pump curves, and for some to a more marked degree than on the graph shown. Such a condition is, in some installations, a disadvantage, especially in cases where the h – Q curve bows upward more than it does in this one.

To illustrate the effect of this bowing, suppose this pump encountered a total lift of 73 feet. The h – Q curve rises higher than that value but drops below it before zero flow is reached. This means there are two points on the h – Q curve that have a head value of 73 feet, and that the pump can discharge either about .25 cfs or almost 1 cfs at that head. The unit will proceed to do just that, alternating back and forth in an uncertain manner called "hunting." Under such flow conditions the user of the water has considerable difficulty controlling the discharge stream, which makes the unit undesirable at this critical head. It is obvious, if the upward bow from zero flow to the right is higher and longer than that shown, that there is a greater zone of performance affected by this hunting characteristic.

We have no way of comparing the sizes of the pumps whose characteristics are pictured in these last four graphs, but it is probable that the screw pump is in the smallest case, the mixed-flow next, and the centrifugal the largest. This means the mixed-flow could probably be applied to a smaller well than the centrifugal type and produce a desired flow, while the screw type could be accommodated in a still smaller casing and produce the same discharge as the other two. Deep-well turbines are the actual subject of the preceding comparison.

A note of caution should be sounded here regarding the strict acceptance of these so-called typical pump curves as truly typical. Great variations can be found in the characteristics of a given type (screw or centrifugal), and when these two are mixed the results are that much more diverse. Curves for the centrifugal and screw-type are more "typical" than that for the last one shown for the preceding reason. You should not necessarily use these curves as a basis for deciding the type of runners responsible for other sets of characteristic curves. In fact, the type is not so important as the shapes of the curves themselves, which indicate what the pump can do and how it will accomplish it.

Application of pump characteristic curve relationships

It would be a mistake not to note and be able to put to use the fact that the five curves—h – Q, over-all efficiency Q, input h.p. – Q, kw-h. per hour input Q, and w.h.p. – Q—are all interrelated in one way or another.

Actually these relationships have been put to use several times in the previous discussion, and completely in drawing the pump characteristic curves. The value in understanding these relationships lies in the fact that often only part of the data

are available, but one can build up the rest by starting with what is provided. For example, given Q and h for a given pump from equation 5 or 6 (whichever is applicable), you can obtain the water horsepower. Usually the input horsepower can be obtained at the electric meter,* while the given Q is being delivered. With input h.p. available and w.h.p. computed, we obtain over-all efficiency using equation 8 where per cent over-all efficiency = $\frac{\text{output h.p.}}{\text{input h.p.}} \times 100$. With input h.p. measured, the input kw-h. per hour is input h.p. $\times$ .746.

The immediately preceding steps are the normal path in finishing these data. Suppose in contrast the over-all efficiency Q and the input h.p. to be fixed for a series of Q values. You can compute the over-all head against which the pump can operate as follows:

$$\text{First, over-all efficiency (known)} = \frac{\text{output h.p. (unknown)}}{\text{input h.p. (known)}} \times 100 \qquad \text{eq. 7}$$

$$\text{Output h.p.} = \frac{\text{input h.p.} \times \text{over-all efficiency}}{100} \text{ (This will be a fixed value for each Q value)}$$

$$\text{Then, output h.p.} = \frac{Q \times \text{pounds/unit volume} \times \text{lift in feet}}{550 \text{ or } 33{,}000 \text{ (whichever fits)}} \qquad \text{eq. 5 or 6}$$

$$\text{Or, lift in feet} = \frac{\text{output h.p.} \times 550 \text{ or } 33{,}000 \text{ (whichever fits)}}{Q \text{ (gpm or cfs)} \times \text{pounds/unit vol.}}$$

This is the general form of the equations. Typical solutions may help to make them clearer.

Problem 1: Suppose for Q = 1.3 cfs, the over-all efficiency was 52 per cent and the input h.p. was not to exceed 27.5. We wish to know the maximum over-all head that this pump could operate against.

Solution: w.h.p. or output h.p. $= \frac{\text{input h.p.} \times \text{over-all efficiency}}{100}$

$$= \frac{27.5 \times 52}{100} = 14.3 \text{ h.p.}$$

$$\text{Total lift in feet} = \frac{\text{output h.p.} \times 550 \text{ (Q is in cfs)}}{Q \times 62.4 \text{ (Q is in cfs)}} = \frac{14.3 \times 550}{1.3 \times 62.4} = 96.95 \text{ feet}$$

Problem 2: A pump is to be bought to deliver water through an over-all lift of 77 feet, and the motor driving it should not be over 20-h.p. nor use more than that amount, because the current supply from a local plant was limited. Sixty per cent over-all efficiency is guaranteed by several pump manufacturers. How many gpm can the pump deliver?

Solution: w.h.p. $= \frac{\text{input h.p.} \times \text{over-all efficiency}}{100} = \frac{20 \times 60}{100} = 12$ w.h.p.

$$\text{Total lift} = \frac{\text{output h.p.} \times 33{,}000 \text{ (Q is in gpm)}}{Q \times 8.33 \text{ (Q is in gpm)}}$$

$$77 = \frac{12 \times 33{,}000}{Q \times 8.33} \text{ or } Q = \frac{12 \times 33{,}000}{77 \times 8.33} = 617.4 \text{ gpm}$$

Problem 3: A pump bought to deliver 875 gpm at a total lift of 146 feet and with an over-all efficiency of 61 per cent is running with a 40-h.p. motor directly connected to it. Why does the motor overheat?

* Note: This procedure will be discussed later.

Solution: From equation 6, w.h.p. $= \frac{875 \times 8.33 \times 146}{33{,}000} = 32.24$ output h.p. or w.h.p.

Over-all efficiency $= \frac{\text{output h.p.} \times 100}{\text{input h.p.}}$ or input h.p. $=$

$$\frac{\text{output h.p.} \times 100}{\text{over-all efficiency}} = \frac{32.24 \times 100}{61} = 52.85$$

The reason the 40-h.p. motor was running too hot was that it was overloaded.*

The preceding illustrates what has been stated before. It is possible to develop missing characteristics of pumps if certain others are available.

* Note: This overload is not so large as it appears on casual inspection. A motor delivers its rated h.p. or brake h.p. to the shaft at 100 per cent load. If this motor were 91 per cent efficient, it would draw $\frac{40}{.91} = 43.96$ h.p., or the input h.p. would be 43.96 h.p. at rated capacity. For this reason 52.85 h.p. input is $52.85 - 43.96 = 8.89$ h.p. overload on the input (about 22 per cent overload, which could cause serious heating). Such motors can usually be run with 10 per cent overload without injury, which would mean 44 brake h.p. and 48.36 input h.p. at that overload.

SELECT THE PUMP THAT MEETS YOUR NEEDS

Consider these factors

When installed, a pump has to meet certain requirements. A particular discharge rate is wanted, and a given lift must be overcome. You may wish to have the discharge rate stay within certain limits while the total lift varies a given amount. The former situation is more easily met than the latter, when the changes in total lift that must be met may make the limits set for discharge flow physically impossible to accomplish. When the desired limits cannot be met, a compromise has to be arrived at to adapt the equipment as well as possible to the actual situation in the field. The curve that controls the pumping cost is the over-all efficiency (equation 8).

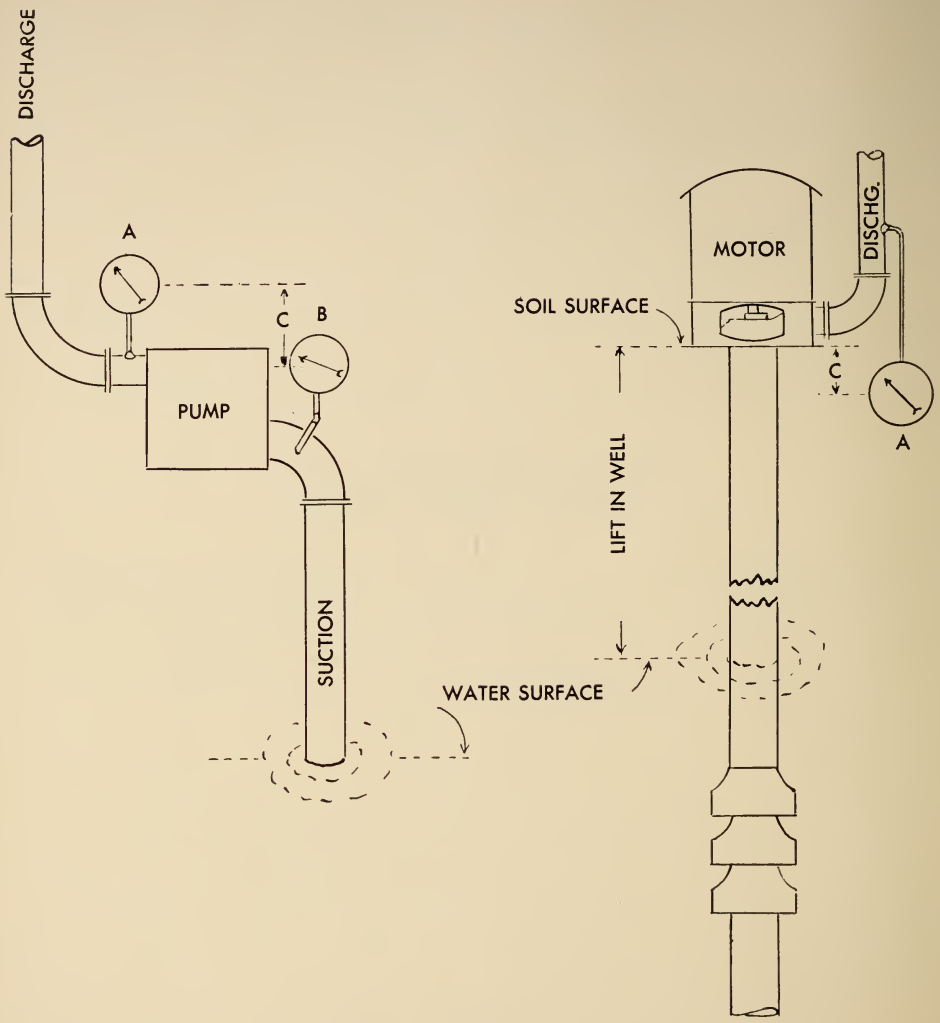
Keep this in mind when deciding on a pump to meet your set of conditions: Select the unit that will meet the situation while operating at the highest efficiency or over the highest range of efficiencies.

Earlier discussion has covered the term *total pumping lift*, which can be a combination of suction lift and discharge lift in the case of the horizontal centrifugal, and is normally assumed for deep-well turbines and other vertical-shaft pumps to be the total lift from pumping-water levels to the surface of the ground, plus the discharge lift above ground surface (See drawing, page 26).

For either the horizontal centrifugal or the deep-well turbine, the pumping-water levels and the discharge levels may vary in the field for several reasons. As a result the total pumping lift can be expected to fluctuate, probably during the pumping season or possibly from season to season. All of the h – Q characteristic curves have a tendency to show a decrease of Q with increasing head for all pump types. We must generally consider this fact and decide that at the maximum expected head some minimum flow is to be produced by the proposed pump.

This fixes one point on the h – Q curve, and the flows produced at the lowest expected total lift will depend on the shape of the h – Q curves for the respective manufacturer's products. Now the over-all efficiency curves become important. Perhaps two or three pumps are reason-

ably close to the desired demands of h – Q, so far as meeting the field conditions is concerned. Inspection of the over-all efficiency curves may demonstrate that only one or maybe none is meeting the range of conditions efficiently.

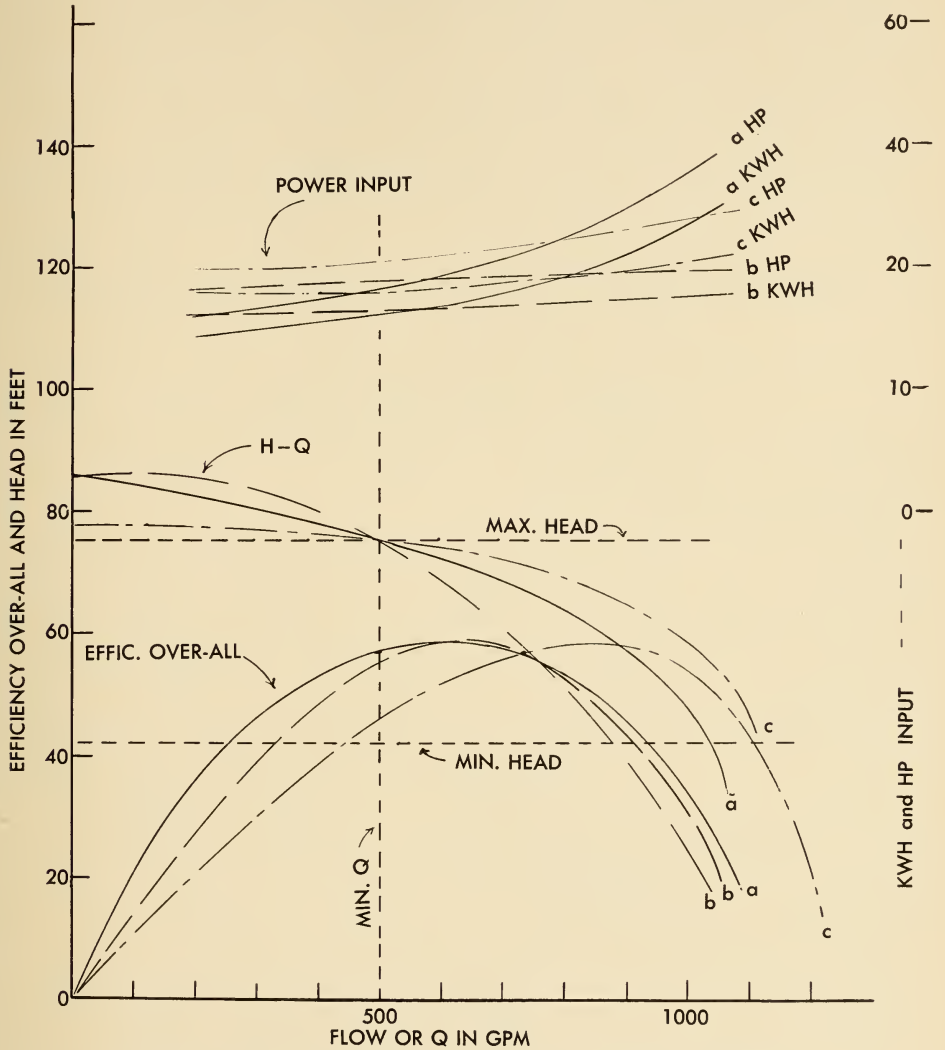


Horizontal centrifugal and deep-well turbine drawn roughly to compare and contrast methods of measurement of total lift for each.

A typical problem

Suppose a centrifugal pump is to take suction from a source of water that is subject to natural changes in elevation, and also that the pipe line it discharges into is opened at various intervals in irrigating so that the discharge lift changes. Assume the maximum expected lift would be 77 feet and the minimum 42 feet. Also suppose the operator of this pump must have at least 500 gpm to irrigate satisfactorily.

The owner has specified these two limits, 500 gpm at 77 feet total lift, and has given several manufacturers' representatives this specification as well as the minimum total lift of 42 feet, requesting specifically that efficiencies and input power data be supplied as for field conditions (true over-all efficiency or input power values). The owner then gets back some tables and some characteristic curves, which he assembles as a group of



Characteristic curves drawn for 3 pumps, a, b, and c, for comparison.

characteristic curves and a table to see how they meet his problem. The illustration above gives the completed graph he produced, and the tables that were supplied and computed for three pumps are shown on page 28.

The curves on the graph indicate that the operator will get his desired flow (500 gpm) when the total lift is 77 feet because all three (pumps, A, B, C) h – Q curves pass exactly through h = 77 feet at Q = 500 gpm. Flows for the respective pumps will vary considerably at the minimum head of 42 feet, as shown at the top of page 28.

Pump	Discharge (gpm)
A	1,040
B	880
C	1,120

This means the discharge from the pumps will change quite a bit from 77-feet head to 42-feet head, and it appears probable that Pump C would supply him with the greatest amount of water at the lower head. Consequently the total pumping time might be shortened and a saving made by using this pump (actually the h – Q curve for Pump C is above that for both A and B between heads 42 and 77 feet, which means Pump C can deliver more flow at any head between those two heads).

This argument might be sufficient to cause the operator to decide on Pump C, but we can study the situation a little more closely by considering the effect of the over-all

Tabulation of Pump Characteristics As Supplied and As Computed

Characteristic in question	Pump A		Pump B		Pump C	
	Supplied data	Computed data	Supplied data	Computed data	Supplied data	Computed data
Q in gpm	250	.....	250	.....	250	.....
Head in feet	81.5	.....	85	.....	77.2	.....
Over-all efficiency (per cent)	42.0	.....	33.0	.....	.....	25
Output h.p.	.....	5.14	.....	5.36	.....	4.875
Input h.p.	.....	12.34	.....	16.25	19.50	.....
Input kw-h. per hour	.....	9.2	.....	12.12	14.55	.....
Q in gpm	500	.....	500	.....	500	.....
Head in feet	75	.....	75	.....	75	.....
Over-all efficiency (per cent)	57.5	.....	55.2	.....	.....	46
Output h.p.	.....	9.47	.....	9.47	.....	9.47
Input h.p.	.....	16.46	.....	17.15	20.60	.....
Input kw-h. per hour	.....	12.28	.....	12.79	15.36	.....
Q in gpm	750	.....	750	.....	750	.....
Head in feet	66.5	.....	55	.....	71	.....
Over-all efficiency (per cent)	56.1	.....	56.5	.....	.....	57.0
Output h.p.	.....	12.59	.....	10.42	.....	13.45
Input h.p.	.....	22.43	.....	18.45	23.60	.....
Input kw-h. per hour	.....	16.74	.....	13.76	17.60	.....
Q in gpm	1,000	.....	1,000	.....	1,000	.....
Head in feet	48.0	.....	24.5	.....	58.4	.....
Over-all efficiency (per cent)	34.2	.....	31.5	.....	.....	54.0
Output h.p.	.....	12.11	.....	6.185	.....	14.75
Input h.p.	.....	35.4	.....	19.63	27.31	.....
Input kw-h. per hour	.....	26.4	.....	14.65	20.39	.....

efficiency on the input h.p. or input kw-h. Sometimes it is cheaper in the long run to pump a little longer at a higher efficiency overall than to pump the shorter time at a substantially reduced efficiency, because it is the gross consumption of electricity that affects the power bill. The power bill on a pumping load is considered at length later.

We need to make a few more assumptions (it can be assumed you have made those that follow before you finally select your pump) before undertaking a study of the effect of varying discharge rates and over-all efficiencies (both affect h.p. input and kw-h. per hour use).

Assume that the pump will have to operate 1,000 hours per year if the minimum discharge of the pump is delivered continuously. Obviously the minimum flow will be only a transient situation in the field, but this assumed flow is a starting point. Next, we can assume that the pump will operate ¼ the time at 77-feet lift, ¼ the time at 42-feet lift, and ½ the time at a lift midway between 77 and 42 feet, or 59.5-feet lift. This is a rough approximation of what may actually be the distribution of pumping lifts through the season. The discharges provided by the respective pumps are defined by these three heads where the h – Q curves cross them. The h – Q data for the three pumps can be picked off the curves as shown below.

Discharge in Gallons Per Minute

Pump	Head operated against in feet		
	77.0	59.5	42.0
A	500	870	1,040
B	500	720	880
C	500	990	1,120

The over-all efficiencies and kw-h. per hour input picked from these same curves for the three pumps operating at these heads are as follows:

Over-all Efficiency and kw-h. Input Respectively

Pump	Head operated against in feet					
	77		59.5		42	
	Efficiency	kw-h.	Efficiency	kw-h.	Efficiency	kw-h.
A	57.5	12.28	49.0	20.4	42.0	28.7
B	55.2	12.79	58.0	13.5	46.0	14.0
C	46	15.36	55.0	20.4	40.0	21.0

The total gpm delivered per year on the assumption it takes 1,000 hours of pumping at 500 gpm will be: $1{,}000 \times 60 \times 500$ gallons.

Any given pump, it has been assumed, will pump ¼ of a certain period of time at the low rate, ½ of that time at some intermediate rate, and ¼ the time at the maximum rate. We can let this unknown total time = t (hours). Then, for Pump A, $(\frac{1}{4}t \times 60 \times 500) + (\frac{1}{2}t \times 60 \times 870) + (\frac{1}{4}t \times 60 \times 1{,}040) = 1{,}000 \times 60 \times 500$

Clearing this equation, $(\frac{1}{4}t \times 500) + (\frac{1}{2}t \times 870) + (\frac{1}{4}t \times 1{,}040) = 1{,}000 \times 500$
or $125t + 435t + 260t = 1{,}000 \times 500$
or $820.0t = 500{,}000$
and $t = \frac{500{,}000}{820} = 610$ hours (total pumping time using Pump A)
So $\frac{1}{4}t = 152.5$ hours, and $\frac{1}{2}t = 305.0$ hours.

By the same procedure for Pump B,
$(\frac{1}{4}t \times 60 \times 500) + (\frac{1}{2}t \times 60 \times 720) + (\frac{1}{4}t \times 60 \times 880) = 1{,}000 \times 60 \times 500$
$t = 710$ (total pumping time using Pump B)
$\frac{1}{4}t = 177.5$, and $\frac{1}{2}t = 355.0$ hours

For Pump C,
$(\frac{1}{4}t \times 60 \times 500) + (\frac{1}{2}t \times 60 \times 990) + (\frac{1}{4}t \times 60 \times 1{,}120) = 1{,}000 \times 60 \times 500$
$t = 555.5$ (total pumping time using Pump C)
$\frac{1}{4}t = 138.87$, and $\frac{1}{2}t = 277.74$ hours

It is evident from the preceding that the three pumps will each require a certain period of time to pump the required amount of water and that B will consume the longest time to complete the job. Each will consume a certain amount of power at each of the three pumping rates; the calculations for the total power demand follow.

Pump A:	**High rate**	**Intermediate rate**	**Low rate**	**Total time**
Time used, hours	152.5	305.0	152.5	610 hr.
kw-h. per hour	28.7	20.4	12.28	
Total kw-h. at each rate	4,376.75	6,222	1,872.7	
Final total kw-h. used				12,471.45

Pump B:	**High rate**	**Intermediate rate**	**Low rate**	**Total time**
Time used, hours	177.5	355.0	177.5	710 hr.
kw-h. per hour	14.0	13.5	12.79	
Total kw-h. at each rate	2,485	4,793.5	2,270.22	
Final total kw-h. used				9,548.72

Pump C:	**High rate**	**Intermediate rate**	**Low rate**	**Total time**
Time used, hours	138.87	277.74	138.87	555.5 hr.
kw-h. per hour	21.0	20.4	15.36	
Total kw-h. at each rate	2,916.27	5,665.9	2,133	
Final total kw-h. used				10,715.17

Returning to the curves on the graph, we find that the pumps A, B, and C will require different motor sizes. A needs a 40-h.p. motor; B needs a 20-h.p. motor; and C a 30-h.p. motor.

This situation in itself indicates there may be a preference in Pump B because it is going to have the lowest first cost, and the preceding calculations indicate this pump will be cheapest to operate because it uses the least power in doing the job. The power alone* will cost about 1.25¢ per kw-h.; then the power used will cost as follows:

* Each of these pumps will have an additional fixed power cost added to the power-use charges as follows:
Pump A, 40-h.p., will cost an additional $200 per year for demand charge.
Pump B, 20-h.p., will cost an additional $100 per year for demand charge.
Pump C, 30-h.p., will cost an additional $150 per year for demand charge.

Pump	Power used, kw-h.	Cost each year at 1.25¢ per kw-h.
A	12,471	$155.89
B	9,549	119.36
C	10,715	133.94

These charges, which are part of the figuring of a power bill, will be taken up in detail later.

The above data indicate that Pump B will be cheaper to operate than the other two, so far as power charges are concerned. One factor has been left out in the consideration of costs of operation—that is the labor costs. Pump B runs 100.5 hours longer than Pump A, and 156.5 hours longer than Pump C. If labor costs $1.00 an hour it appears that Pump B is still the cheapest. If, on the other hand, labor exceeds $1.00 an hour one of the other pumps may be the cheapest.

There is another (perhaps clearer and simpler) way of figuring the time required by these three pumps to do the annual irrigating job. If the maximum and minimum discharge each use ¼ the pumping time and the intermediate flow ½ the pumping time, it is possible to set up the following relationships to obtain the average flow for the whole time.

General relationship:

¼ max. flow + ½ intermediate flow + ¼ min. flow = average flow

For Pump A:

$(\frac{1}{4} \times 1{,}040) + (\frac{1}{2} \times 870) + (\frac{1}{4} \times 500) = 820$ = average gpm from Pump A

But to do the job, if the flow is 500 gpm, takes* 1,000 hours per the original assumption; therefore

$500 \times 1{,}000 = 820 \times x$ (hours) and $x = \dfrac{500 \times 1{,}000}{820} = 610$ hours (operating time for Pump A)

This solution is almost identical to that used previously except that the value t is left out of the solution in this simpler form. Since the solution closely follows the previous one and the answer for operating time is found to check for Pump A, it will not be repeated for the other two pumps.

* Note: From a table located later in this discussion, 500 gpm flowing 1,000 hours would put on 1,111 acre inches. This flow rate, 500 gpm, might have been obtained if the total depth of irrigation per year and the acreage covered had been given thus: 55 acres irrigated 20.2″ deep would use 1,111 acre inches, and 500 gpm would take 1,000 hours to accomplish this from these relationships 500 gpm $= \dfrac{500}{450} = 1.111$ cfs; 1 cfs. = 1 acre inch per hour. Then to put on 1,111 acre inches at the rate of 1.111 cfs or 1.111 acre inch/hr. would consume $\dfrac{1{,}111}{1.111} = 1{,}000$ hours.

What capacity pump do you need?

Under the previous discussion it was assumed that a certain pumping time per season was necessary at a given discharge rate. This was based on supplying some fixed depth of water to the area being irrigated each irrigation season. It is general practice to have between 40 and 80 acres under each cubic foot per second of water supply. The basis for this practice rests on the following reasoning (and on basic data in the table on page 33).

The sun can extract (by evaporation and transpiration) about .2 inch depth of water per day from the soil and crops during the summer months in most of the agricultural areas of the interior valleys of California. This means that there will be $30 \times .2$ or 6 inches of water extracted per month; this is the amount the irrigation must replace if the crop is to be kept supplied adequately in one of these valleys.

One cfs serves 40 to 80 acres. Assume that 80 acres in the northern part of the San Joaquin Valley are being studied to determine what supply is needed or what size pump is required. If the evaporation-transpiration from this field is at the normal rate of .2 inch per day or 6 inches per month, 480 acre-inches will be required per month to replace these losses. If 1 cu. ft. per sec. is available, it will take 480 hours' pumping to supply the required water. There are 24 hours per day and 30 days in the month, or a total of 720 pumping hours available. $\frac{480}{720} = \frac{2}{3}$, or only two thirds of the gross time is apparently used for pumping.

Actually a longer time than the net 480 hours will be required of the pump because the water will be lost by evaporation and by seepage in the ditches, by seepage below the root zone in part of the fields (which is waste), and by direct waste often caused by escape from the tract entirely. It is because these extra requirements are put on the pump to make up for losses that the practical load per cubic foot per second supply is assumed to be 80 acres. There are other minor reasons, such as the desirability of not being tied to the irrigation work completely throughout the months and the provision of time to shift the flow from one field to another. In reality, to put 80 acres under each cubic foot per second supply is often a mistake—60 acres per cu. ft. per sec. is the more workable compromise. (Desert areas often have 1 cu. ft./sec. for 40 to 50 acres, while in coastal areas as many as 100 to 120 acres per cu. ft./sec. are easily possible.) This 60 acres is an average figure between the 40 and 80 acres per cubic foot per second specified earlier for the interior valleys.

Use a reservoir with a small water supply

The capacity of pump required for a given area is in proportion to this general ratio if the supply is more or less unlimited. In cases where the supply is limited by the capacity of its source, an expedient is possible in the form of a reservoir. With storage of the pumped supply, it becomes feasible to pump continuously at, say, ½ cubic foot per second, or 225 gpm; then to store the flow for 12 hours in the reservoir; and finally to draw 1 cfs from the reservoir, while the pump continues to pump for the other 12 hours each day.

This does not give the 480 hours irrigation time required per month per the previous solution for an 80-acre farm with 1 cfs flow, but 12×30, or 360 hours. It does produce all the water possible from the source and uses it at convenient times and at higher rates, making possible satisfactory coverage of the area it can serve. In this case, probably about $\frac{360}{480} \times 80$ acres could be served, or 60 acres maximum under the conditions specified for the 80-acre area previously studied. To use this flow for 40 acres or less would probably be better judgment.

Rates of application of water for various soil types and methods

Various irrigation methods are available for the application of water to crops. Some of the methods control the rate of application of water to a degree, and soil type exerts an effect on all methods of water application. The following table is a generalized summary of the available data on irrigation water application rates for several main soil types and for four common types of irrigation. It is not intended to be restrictive in its application, but is a general guide permitting a considerable departure from the values given in the table on page 33.

Wide departures are possible from the values given in the table; hence it must be considered a guide and by no means a strict set of rules to go by. The sun is the other factor that largely controls the use or rate of use of the water and thereby de-

Basic Data on Water Requirements of Various Irrigation and Soil Types

Irrigation type	Slope land per cent	Coarse sandy soils		Light sandy loam		Medium silt loam		Clay loam soils		Very heavy clay soils	
		Q per unit	Length of run	Q per unit	Length of run	Q per unit	Length of run	Q per unit	Length or run	Q per unit	Length of run
Basins*	0–2	20 cfs per acre		7.5 cfs per acre		5 cfs per acre		3 cfs per acre		2 cfs per acre	
Borders* or Checks	0–2	1.5 cfs per 10′ width	220′	.75 cfs per 10′ width	440′	.5 cfs per 10′ width	550′–880′	.33 cfs per 10′ width	660′–880′	.3/10′	1,000′
Furrows	0–2	.02 per each	220′	.01 per each	330′	.01 cfs per each	440′ 660′	.008 cfs each	660′	.005 per each	880′
	2–5			.005 each	220′	.005 per each	220′ 440′	.003 per each	440′	.003 per each	550′
	5–8					.002 per each	110′ 220′	.001 per each	330′	.001 per each	330′
Sprinkling	0–2	2″ per hour		.75″ per hour		.5″ per hour		.2″ per hour			
	2–5	2″ per hour		.75″ per hour		.5″ per hour		.2″ per hour			
	5–8	1.5″ per hour		.5″ per hour		.4″ per hour		.15″ per hour			
	8–12	1.0″ per hour		.4″ per hour		.3″ per hour					

* The range in slope 0–2 per cent is in itself a very rough picture of field practices where the actual slopes, particularly with borders, tend to be closer to .2 or .3 per cent rather than this higher limit 2 per cent.

termines how frequently the application need be replaced.

As previously noted, the sun extracts about .2″ of water per day, by evaporation and transpiration in the summer months in most of the interior valleys of the state. Crops having shallow rooting systems draw upon a limited supply of stored water compared to those growing with deep-rooting habits in the same type of soil. Similarly, a crop like alfalfa, which roots 6 to 8 feet or more deep, if grown in sandy soil has much less storage water to draw upon than if grown in a heavy loam or a clay, because the sand cannot store as much water as the clay. For these reasons, though the sun's extraction may be at a constant rate at any time and place, the soil storage may vary widely. Therefore the frequency of irrigation has to be adjusted to the soil type and crop as well. (For further details, see Extension Circular 50.)

The applications of the table are as follows:

Problem 1: Basin irrigation—Suppose it has been decided to block out an orchard into 1-acre units for irrigation. Suppose further that the soil is very heavy clay in one area and light sandy loam in another, and the grade within the orchard is not over 1 per cent in either east-west or north-south directions (between 0–2 per cent). The lighter soil (sandy loam) will require a flow of 7.5 cfs per block (per acre) and the very heavy clay only 2 cfs per block. The pump supply must be at least 7.5 cfs for the sandy loam area, and the stream will have to be divided into 3-plus blocks to irrigate the heavy clay area, but 7.5 cfs must be supplied.

Problem 2: Suppose coarse, sandy soil and a clay loam soil were being irrigated from the same pump by sprinkling, and the lengths of sprinkler run happened to permit the coverage of one acre each. What size pump would be needed if the slope on the coarse, sandy soil was between 0 and 2 per cent, and on the clay loam between 5 and 8 per cent?

From the table for 0–2 per cent on the coarse, sandy soil, 2 inches per acre per hour is permissible, and for the clay loam with 5 to 8 per cent slope only .15 inch per acre per hour. If a pump large enough to supply 2 acre inches per hour to the coarse, sandy soil were bought, it would have a capacity of 2 cfs or 900 gpm. The same pump would supply $\frac{2}{.15}$ as many sprinklers on the clay loam area or 13⅓ times as many one-acre runs.

If the pump is to run as efficiently when serving the clay loam as for the coarse sand, the owner will have to supply at least 12 more sprinkler lines to operate all at once in order to use the 900 gpm which the pump can safely deliver to the coarse sand in one run. This is an unusual situation, but it illustrates the range of problems that may be met in deciding what size pump to buy.

Problem 3: Borders and furrows are in a sense respectively wide and narrow ditches down which the water is led. Hence, in the case of the borders, the flow required depends on how wide they are in order that the supply can spread over it evenly. (The table states this in flow, cfs, per 10 feet of width.) The furrows are much alike and the flow per furrow can be stated. Thus if 1 cfs were available one could have borders 20 feet wide on a medium silt loam, and they could be 550 to 880 feet long for 0–2 per cent slope. Furrows on the same soil but with 5–8 per cent slope would need only .002 cfs each, and should be only 110 to 220 feet long. It would take 500 furrows all operating at once to use 1 cfs. It is common practice to start the flow down borders and furrows at a high rate, then cut back the delivery so that a continuous supply is afforded the length of the strip, and in the case of borders also the breadth for the remainder of the irrigation.

It is clear that the size of pump depends, first, on how many acres are to be served; second, on how much water is to be put on per acre per irrigation season;

and third, on some fixed operating period of days or hours. All this is based on the assumption that the supply and correspondingly the size or capacity of the pump are unrestricted.

Unfortunately this is not generally the case. The well, where this is the source, is the limiting factor in supply. The pump must fit the well capacity. By the same reasoning the well capacity controls the area that can be adequately irrigated. The method of irrigation and the soil type necessitate modifications in the irrigation program to permit the efficient application of water, but the crop requirements must be met whatever the procedure adopted.

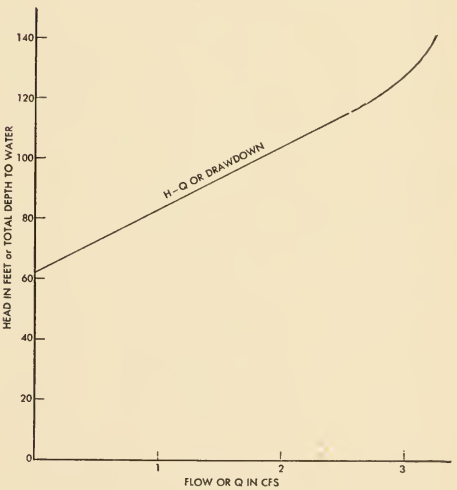
Well characteristics

A well is different from a stream or pond only in the matter of visual evidence of the supply. In the latter two sources, if the water is not there it is obvious, and also if the pump uses the water from a stream or pond faster than it is supplied it can be noted by a lowering of the supply water levels. The same thing goes on in a well, but it is not so apparent. In fact, a well has a characteristic h – Q curve for any given time and, if ground-water levels are changing, a series of h – Q curves follows the general change in ground-water levels for the area as they move.

Well-characteristic curves. The accompanying graph illustrates a typical characteristic curve for a well. Several things shown on this curve need further comment. Over at the left margin where Q = zero the h – Q or drawdown curve touches 62 feet head. This means when there is no pumping the water in the well (static water) is 62 feet down from the measuring point (presumed to be the surface of the ground). Out to the right on the h – Q curve near Q = 2.5 cfs the curve begins to turn upward, or h begins to increase faster with increasing Q than had been the case. This means the first underground stratum (maybe the only one) has begun to be uncovered by the receding water. In other words, the surface of entrance to the well (the supply surface) is being diminished, so that the head increases faster than it had been.

The body of the h – Q curve between zero Q and about 2.5 cfs is a straight line up to the point where the first stratum is being uncovered. In this zone h changes in direct proportion to the change in Q. This is a typical h – Q, or drawdown, curve from an artesian well. It is not a flowing artesian well, but as the supply strata are under pressure the top one is covered a good many feet by standing or static water.

Almost all wells in California are of this general type, but the slope of the curve differs as well as the position of the static water measurement (h where Q = zero). Furthermore, the Q scale may be much different from that given in the graph. A nonartesian well has no pressure in the water-supplying zone or zones; hence as soon as water is withdrawn less and less of the supply area is available. As a result the h – Q curves continue an upward trend at an increasing rate throughout their length.



A typical artesian well character or drawdown curve. (First water-bearing stratum down about depth 120')

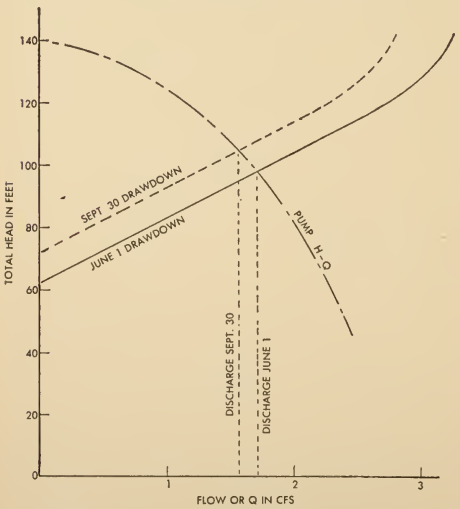
Specific capacity of wells. One notable feature of the artesian well h – Q curve is the straight line section before the exposure of the top stratum occurs. It is possible to evaluate the well in terms of discharge of water per unit of lift (specific capacity). This can be illustrated from the graph as follows:

Head	Q
62′	0.0 cfs
104′	2.0 cfs

Difference in h and Q for two points gives h = 42 feet and Q = 2.0 cfs or 900 gpm. Specific capacity $= \frac{2.0}{42} = .0476$ cfs per foot drawdown of $\frac{900}{42} = 21.41$ gpm per foot drawdown. Or,

Head	Q
80	.86 cfs
114	2.49 cfs Diff. or h and Q = 34

feet for 1.63 cfs, or 734 gpm. Specific capacity $= \frac{1.63}{34} = .048$ cfs per foot drawdown (error due to failure to read points on curves accurately) or $\frac{734}{34} = 21.58$ gpm per foot drawdown. (A slight error in reading the data from the well characteristic curve causes this value, 21.58, instead of 21.41 gpm per foot drawdown above.)



Same drawdown curve as on page 35 (June 1) plus drawdown expected for Sept. 30 and including a pump h-Q curve.

The specific capacity of this well in the straight portion of the curve is about 21 or 22 gpm per foot drawdown, which is about an average figure. Many wells produce at higher and others at lower rates. In general, wells are used so that the production (Q) falls safely within the straight section of these h – Q curves, or somewhat to the left of the point where the effect of uncovering the first stratum is felt.

Changing water levels in wells. The curve (at left) might characterize this well on June 1 of some year. By September 30 of the same year the static water levels might have lowered 10 feet, which would cause the straight part of the curve to move upward 10 feet. The first stratum will still be exposed in this well at a depth of about 116 feet; thus by September 30, when the drawdown curve had shifted upward 10 feet (see graph below), it would cross the 116-foot depth at Q = 2.1 cfs (the drawdown curve for June 1 had crossed the 116-foot depth at Q = 2.6 cfs).

The effect of the seasonal lowering of water tables has been to lower the capacity of the well at this critical point, which illustrates what has been stated earlier—that one should require less flow of a well than the amount that will cause the uncovering of the top water-bearing stratum. In a sense, the limit of the well is reached there, and if water tables go down the Q limit becomes smaller. Fortunately this uncovering of strata does not enter into the consideration of most well-characteristic curves because the top stratum is sufficiently deep to eliminate it as a factor.

The preceding indicates that a well has a fixed h – Q curve at a given time and that a pump also has a fixed h – Q characteristic at all times. It is logical to conclude then that, when a pump and a well are put together, a fixed flow results, because somewhere an over-all head is ar-

rived at that is agreeable to both pump and well and that determines the flow. This is actually the case. All one needs to understand is how this "agreeable" head is reached.

Fitting pumps and wells

The simplest situation. The easiest situation to meet in putting a pump in a well, if the well-drawdown curve is known, is first to have only that curve to deal with (no changing underground water levels) and second for the pump to discharge the water at the surface of the ground at the top of the well.

Such a simple installation can be assumed using the lower drawdown line (marked June 1 drawdown on the graph) and assuming for the present that it is to remain steady throughout several years. We need also to assume that the pump discharges into a ditch or the equivalent at the surface of the ground; therefore the total lift corresponds to the drawdown curve head for any discharge rate. If a given pump h – Q curve is plotted on this same figure, it will cross the well-characteristic curve at some point. This intersection is the discharge of this pump from this well until conditions change in the well or until another pump is put in.

Now if we suppose that conditions do change by September 30 so that the gross lift in the well is as shown, a new drawdown line, also shown, represents the behavior of the well, and it intersects the pump h – Q curve at a new point. In this case, by September 30 the well-water levels had gone down, which meant the total lift had gone up. Thus the intersection with the pump h – Q curve is at a higher point than for the June 1 conditions. Higher heads on these pumps imply in general lower flow rates, and this is true in the graph used. These are relatively simple conditions for a pump to meet, but so far as the lift in the well is concerned this figure illustrates the whole problem.

More complex situations. Pumps do not always discharge freely at the surface of the ground at the well. Frequently they discharge into a pipe line of some sort; and the pressure, or head, they operate against is affected by the location of the delivery outlets along the pipe. The total head they operate against is the discharge head plus the pumping lift in the well.

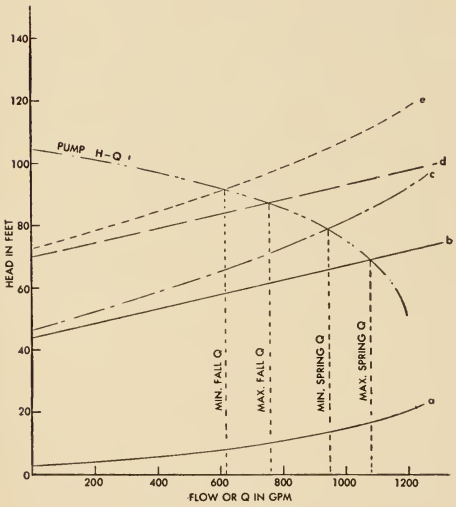
The graph on page 38 illustrates a situation where the water levels in the well vary during the season (curves b and d) and where the discharge head above the top of the well also varies (lowest curve, a, on the graph). A typical pump h – Q curve has been drawn for use in noting the effects on the pump discharge of the variable over-all head conditions it has to meet. The lowest head for the pump can be assumed as the total lift in the well for any flow and at any season (in other words along the well-drawdown curves).

Altogether, four intersections occur between the pump h – Q curve and the two well-drawdown curves plus the two maximum lift curves, c and e. These latter two curves are the result of adding the discharge lift values for any given flow (bottom curve in the graph) to the corresponding point on the two seasonal drawdown curves. For example, at Q = 800 gpm the highest discharge head is 10 feet; and adding this to the spring drawdown value at 800 gpm (which is 65.5), the maximum lift for 800 gpm must be 75.5 feet. These two maximum lift curves show the ultimate head required of the pump for the two seasons. The discharge possible at the four intersections with the pump h – Q curve represents maximum and minimum discharge rates for each season; they are so labeled on the graph.

The situation pictured is typical of many pumping plants and becomes a part of the solution when a pump is selected. It obviously makes the solution more complicated than the one covered earlier (page 36), but the procedure is identical in both. One more complication is often a part of these problems. Up to this point,

the figures have shown wells having fixed spring and fall drawdown curves. In many areas there is a continuous change (generally lowering) of water levels, and so on the graphs the drawdown curves keep shifting (upward if the total lift in the well increases) year by year.

Under such circumstances we must project the probable future drawdown curves onto the graph on which the pump characteristics are to be studied, then correct the drawdown curves for the variations in discharge lift above the top of the well (if any). From the corrected curves we learn the probable maximum pumping lift in, say, 5 years, and also the minimum, and can, first, specify more closely what is desired of a pump and, second, determine whether the pumps described in the bids meet the requirements, and which pump is best.



More complex situation in well and on discharge line from pump illustrated, to show effect on pump discharge using given pump h-Q curve. Curve a is the head developed on the discharge of the pump with varying flow; Curve b is the minimum pumping lift in the well (drawdown curve for highest water levels); Curve c is the over-all pumping lift (discharge plus lift in well) for minimum-lift conditions; Curve d is the drawdown curve for the well for maximum pumping lift in the well; Curve e is the maximum total lift for the pump (discharge head plus drawdown).

The discharge head curve can be more complex than that given on the bottom of the graph used, where it is a smooth curve rising from left to right. Some of these variables can be illustrated if certain assumptions are made as a base.

Problem 1: Suppose the specific yield of a well is 30 gpm per foot drawdown and that this spring the static water is at 32 feet depth. The lowering of water by fall each year amounts to 12 feet; for the last few years water tables have gone down from spring to spring a depth of 4.5 feet and may be presumed to continue doing so. The pump will discharge into a pipe line across a flat stretch of land, then up a hill to another flat or bench. Irrigation on the first flat will be at a low rate and on the higher bench at a greater rate (details in curve A on graph on page 39). Suppose you want to select a pump that will be effective and most efficient 5 years from this coming fall when the minimum flow is to be 750 gpm. The curves on the graph were drawn as follows:

Drawdown curve this spring—Start with static water (Q = zero) where h = 32 feet, and put a dot there. Then if the specific yield of the well is 30 gpm per foot drawdown, for 1,500 gpm delivered the drawdown will be $\frac{1,500}{30} = 50$ feet. Thus the well will draw down to 32 + 50, or 82 feet. Put a point on the location of h = 82 feet and Q = 1,500 feet, and connect these two points with a straight line. This is the curve B drawn on the graph.

Drawdown curve for spring 5 years later—Water levels can be expected to recede for the next 5 years at the rate of 4.5 feet per year. Then 5 years hence the lift in the well will increase 22.5 feet. Starting at static water this spring at 32 feet, move up the zero Q line (left side of chart) to 32 + 22.5 = 54.5 feet. Do the same at Q = 1,500 gpm where h becomes 82 + 22.5 = 104.5 feet and connect the two points. See curve C on the graph.

Drawdown curve for fall 5 years

later—The spring-to-fall increase in water depth is expected to be 12 feet. Accordingly, add 12 feet to static water and to h at Q = 1,500 gpm for the spring curve 5 years later. Connect the two points, and the fall curve is drawn as D on the graph.

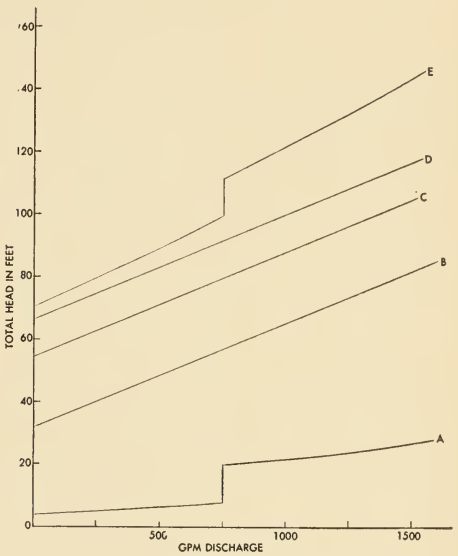
Maximum lift to flat and to bench in fall 5 years hence—Add to fall drawdown curve just drawn the increased pumping head corresponding to zero Q, 250 gpm, 500 gpm, etc., at those flow lines; and after connecting the points derive this final broken curved line, E on the graph.

All of the characteristics of the well and discharge line are now defined on this last broken curve, and the pump must deliver a minimum of 750 gpm at a head of 115 feet plus, in the fall 5 years hence. Knowing the general shape of pump h – Q curves, we might decide to fix on some point on the "maximum lift to bench in fall 5 years hence" curve as the limiting h – Q point for the pump. In this way more than the minimum Q would be available at h = 115 feet.

This decision would be subject to special consideration in the field. Consequently it will be necessary to assume for this discussion that the proposed pump, meeting conditions for delivery to the bench, is acceptable. In all probability the assumption is the normal solution because, if water is to be delivered to the bench, the over-all head will have to meet the situation pictured along that portion of the curve.

You might decide on a Q of 1,000 gpm at a gross head of 122 feet, or on 1,250 gpm at a gross head of 132 feet. In order to be specific the former, 1,000 gpm at 122 feet, is accepted for discussion here. The pump must meet these h – Q conditions in the fall 5 years hence. The pump vendors would be asked to bid on pumps to meet these requirements.

Another specification that might be made is that the over-all efficiency curve for the pump should be flat in the neighborhood of 750 gpm and both sides of this figure as far as possible. The reason for this requirement is to get a pump having good efficiency spread over a wide range of capacities to improve the over-all economy where such a spread of requirements is expected of the pump. The pump vendors can select from a number of models and supply data on one or more that meet the requirements most satisfactorily. Then by comparison of the various pumps offered it is possible to select the one most suitable for the job. The demands upon a pump pictured in the graph (see below) are somewhat unusual but not seriously so. Almost any conceivable combination of head and Q may be encountered somewhere so that each in-



Drawdown curves for well and reasonably complicated discharge-head curve above pump combined to illustrate resulting maximum pumping head. Curve A is the discharge head above the pump under varying-flow conditions (the vertical job over Q = 750 gpm is, as explained in the text, the result of the bench or elevated portion of the land and not a result of this particular flow); Curve B is the spring drawdown curve for the well; Curve C is the drawdown curve in the spring 5 years from now; Curve D is drawdown curve for the well in the fall 5 years from now; Curve E is the maximum pumping lift (drawdown plus discharge) 5 years from now.

stallation of pump to well becomes a special problem in itself.

Why develop a well?

Without going into the question of well development in detail, it must be emphasized that no new pump should go into an undeveloped well. There are several reasons. The first relates to the previous paragraphs, in which specific curves were drawn to picture the well characteristics. These curves come as the result of making a production test on the well. The production test is not complete until clear mud-free water is being pumped at (or considerably above) the maximum expected draught on the well. The well's characteristics change until clear water is being delivered at the maximum draught.

Actually the completed production test or development improves the well, so that the characteristic curves show higher specific yields than at any earlier stage in the process. It is almost impossible to fit a pump accurately to a well that possesses no characteristic curve. If the mud and sand are all cleaned out of the well, the new pump is saved the injury that may result from handling this material. Sometimes the damage to impellers and bearings caused by developing a well with a new pump can cut over-all efficiencies in half. This is obviously a serious situation to an owner because of immediate injury to the equipment, and, from the by-product, high operation charges from that day on.

For the previously listed reasons we recommend that a well test be completed on any well in which a new pump is being installed. The test or development of the well can be made a part of the well-drilling contract. If so handled, the stipulations in the contract should name the necessity of clearing out all mud and sand to produce clean, clear water at some maximum drawndown or depth in the well, or when some specified discharge rate has been reached. One or the other of these limits will have to be given or left to mutual agreement between contractor and owner so that there is an understanding on the subject. The well-development process has been explained in greater detail in Exp. Sta. Circular 404.

Submersible pumps

A special adaptation to the deep-well turbine is the so-called pencil motor, which can be submerged in water. It has a very long body and relatively small outside diameter and is, therefore, suitable for installation in wells.

These motors are put at the bottom of a submersible pump and are small enough in diameter to permit the flow of water between them and the well casing or lining. They are direct-connected to the impeller shaft, and since they are immediately below the bowls a relatively short driveshaft is possible. No oil tubing or guides are necessary in the discharge column leading to the pump head. The only requirement is to run the electrical energy to the motor in sealed conductors so constructed as to prevent power leakage from them where they pass through the well water, and also to provide seals at their junction with the motor-winding terminals.

In some makes of submersible pumps, an oil line is run down to the motor along with the electrical conductor. The short driveshaft is a definite advantage for a turbine pump in a deep well. The small diameter of the motor required for well service introduces for the manufacturer of these products mechanical problems that have been difficult to overcome. Crooked wells that would cause bearing difficulties for surface-driven pumps can sometimes be effectively pumped using the submersible.

COSTS AND METHODS OF INSTALLATION

Consider the power schedule

There may appear to be no correlation between total power used and the cost per kw-h. which broadly speaking is true, but there is a relationship between the total hours of operation, motor size, and the resulting over-all cost per kw-h. The way the power-schedule cost table is laid out,

(Continued on page 43)

How to compute power costs

On earlier pages we used some power cost data to compute the cost of pumping where several pumps were involved and it was necessary to select the most economical one. These data came from generally available tables such as the following:

Cost of electric power (revised schedule, 1950)

Agricultural power schedule. Applicable to agricultural power service required intermittently throughout the year.

Rate:

Size of installation, h.p. (nominal)	Annual demand charge per h.p.	Energy charge in addition to demand. Rate per kw-h. per h.p. per year.		
		First 1,000 kw-h.	Next 1,000 kw-h.	All over 2,000 kw-h.
1– 4.9	$6.62	1.54c	.74c	.53c
5– 14.9	5.56	1.32	.74	.53
15– 49.9	5.03	1.22	.74	.53
50– 99.9	4.50	1.11	.74	.48
100–249.9	3.97	1.11	.74	.48
250–499.9	3.97	1.06	.74	.48

Payments: Demand charge payable in 6 equal monthly installments—May to October. Energy charge payable monthly as energy is used.

Here are four typical power cost computations

These problems will be solved as demonstrations using this table to get the annual power costs for the following situations:

Problem number	Motor name plate h.p.	Hours operated per year	kw-h. per year	Remarks
1	5	750	.....	Motor efficiency, 86 per cent Motor 10 per cent overloaded
2	15	1,600	19,650	.....
3	40	.....	62,650	Motor using 44 h.p.
4	10	2,520	.....	100 per cent load—motor efficiency 89 per cent

Power bill for problem 1:

5 h.p. motor running with 10 per cent overload will deliver 5.5 h.p. to pump shaft. 5.5 delivered h.p. will require $\frac{5.5}{86} \times 100$ input h.p. or 6.395 input h.p. Total kw-h. used for 750 hours = $750 \times 6.395 \times .746 = 3{,}578$ kw-h.

Power charges from table:	
Annual demand charge, \$6.62 per h.p. = 5×6.62 =	\$33.10
Energy charge first 1,000 kw-h. per h.p. = 5,000 kw-h. at 1.54¢ per kw-h., but there are only 3,578 kw-h. total use; hence $3{,}578 \times \$.0154$ =	55.10
Total bill,	\$88.20

Power bill for problem 2:

15 h.p. motor uses 19,650 kw-h. per year.

Power charges from table:	
Annual demand charge, \$5.03 per h.p. = $15 \times \$5.03$ =	\$ 75.45
Energy charge first 1,000 kw-h. per h.p. = 15,000 kw-h. at \$.0122 =	183.00
Remaining 4,650 kw-h. at \$.0074 =	34.41
Total bill,	\$292.86

Power bill for problem 3:

40-h.p. motor uses 62,650 kw-h. per year

Power charges from table:	
Annual demand charge, \$5.03 per h.p. or $40 \times \$5.03$ =	\$201.20
Energy charge first 1,000 kw-h. per h.p. = 40,000 kw-h. at \$.0122	488.00
Energy charge next 1,000 kw-h. per h.p. at \$.0074 kw-h., but there are only 22,650 kw-h. left; therefore $22{,}650 \times \$.0074$ =	167.61
Total bill,	\$856.81

Power bill for problem 4:

10-h.p. motor operating at 100 per cent load for 2,520 hours per year

Total kw-h. per year = $\frac{10}{.89} \times .746 \times 2{,}520 = 21{,}122.7$ kw-h.

Power charges from table:	
Annual demand charge \$5.56 per h.p., or $10 \times \$5.56$ =	\$ 55.60
Energy charge first 1,000 kw-h. per h.p. at \$.0132 or $10{,}000 \times .0132$ =	132.00
Energy charge next 1,000 kw-h. per h.p. at \$.0074 or $10{,}000 \times .0074$ =	74.00
Energy charge next 1,000 kw-h. per h.p. at \$.0053 or $1{,}122.7 \times .0053$ =	5.95
Total bill,	\$267.55

It is possible to determine the net cost per kw-h. for each of these four motor-driven units as follows:

Problem 1: 3,240 kw-h. cost \$88.20 per year or $\frac{\$88.20}{3{,}240} = 2.72¢$ per kw-h.

Problem 2: 19,650 kw-h. cost \$292.86 per year, or 1.49¢ per kw-h.

Problem 3: 62,650 kw-h. cost \$856.81 per year, or 1.367¢ per kw-h.

Problem 4: 21,122.7 kw-h. cost \$267.55 per year or 1.265¢ per kw-h.

the longer a motor operates the larger the gross use of power and the cheaper each kw-h. becomes.

As a result, a small motor running a long time can do a job that a larger motor running much less time could also do. Both might use the same total amount of kw-h. in completing the job, but the small motor because it ran a long time would use power sufficient to move the charges over into the cheaper part of the table. We must remember, though, that labor charges may also accumulate while the small pump works a longer period to produce a given irrigation supply, and that net power savings may be more than counterbalanced by increased labor charges. A reservoir used in conjuction with such a small pumping plant might bring economies in labor.

This power-schedule study illustrates one factor in the selection of pumps and motors that has been omitted until now, namely, that there is an argument for the use of as small a pump as possible so that the power cost can be kept at a minimum. This argument can be supported by the probable lower first cost of a smaller unit, and if a well is involved the smaller pump will discharge at a lower rate. Then the actual pumping lifts will be less than for a larger pumping unit.

These factors all point toward the desirability of keeping the pump sizes as small as possible. We must keep in mind, however, that in order to cover a given area certain capacities are essential to keeping up with the losses by evaporation and transpiration, as well as providing for emergencies and the normal pattern of operation changes—also, that an operator has to have some spare time to himself. The pump size selected is very often a compromise between all of these factors and the labor costs.

Reading the watt-hour meter to check power input: The total power used is accumulated on the watt-hour meter, which the power-company meter readers record at regular intervals. You should occasionally check the power being used at the meter by the pump; there are simple ways of doing this. Each meter set-up has a "constant" or K, which is a factor that can be used in making these checks.

Most meters have a K marked on the face or on the disc, which is the watt hours per revolution of the disc (or the power company can supply the value of K). The kw-h. per hour would be arrived at as shown in the box below.

The table on page 44 gives the disc constants for a group of standard polyphase watt-hour meters handling 220 volts, 3-phase current. If the voltage is 440 instead of 220, the K is double that given in the table. Sometimes extra coupling transformers are inserted into the control panel on large pumping units so that smaller, cheaper meters can be used. In this case, the K is not applicable; for this reason it is best to check with the power company to get the true K for any meter.

$$\text{kw-h. per hour} = K \times \frac{\text{revolutions counted}}{\text{seconds taken for count}} \times \frac{3{,}600\ \text{(seconds per hour)}}{1{,}000\ \text{(watts per kilowatt)}}$$

$$= 3.6 \times K \times \frac{\text{rev. counted}}{\text{seconds taken for count}} \quad \text{and h.p. hours} = K \times$$

$$\frac{\text{revolutions counted}}{\text{seconds taken for count}} \times \frac{3{,}600\ \text{(seconds per hour)}}{746\ \text{(watts per h.p.)}} =$$

$$4.83 \times K \times \frac{\text{rev. counted}}{\text{seconds taken by count}}$$

Disc constants, K, of polyphase watt-hour meters 220-volt—3-phase (K in watt-hours per revolution)

Name and type of meter	Amperes				
	5	10	15	25	50
General Electric:					
D3	1.25	2.5	4.0	6.0	12.5
D6 and D7	1.2	2.4	3.6	6.0	12.0
D14	2.4	4.8	7.2	12.0	24.0
Westinghouse:					
C, OA, OB, RO	1 1/3	2 2/3	4	6 2/3	13 1/3
Sangamo:					
H	5/6	1 2/3	2 1/2	4 1/6	8 1/3

The heading on the table: Amperes-5, -10, -etc. refers to the data on the meter-face plate which identifies its make, type, and capacity in terms of amperes.

The watt-hour meter K is useful in checking the performance of a pump either during a pump test or where it is desired to see how the over-all efficiency is being maintained as the pump ages.

Installing pumping plants

Horizontal centrifugals. It is probable that more horizontal-centrifugal pumps are installed by their owners than are the deep-well turbine types. Precautions must be taken with both, but they are different enough to make it desirable to separate the discussion of the installation into two parts.

Foundations. Horizontal centrifugals, being relatively compact, generally come on a cast-iron base with motor and pump accurately lined up, so that it is necessary only to supply a firm level foundation for the base and to hook up suction and discharge piping. The care taken in these steps pays the owner by retaining in the unit the utmost over-all efficiency incorporated by the manufacturer. For example, the foundation should be firm and level, as previously noted. Do not put in the foundation until you have fixed its location by careful consideration of two factors: (1) the relationship of water supply and position of discharge piping and (2) the desirability of keeping the piping plan as simple as possible.

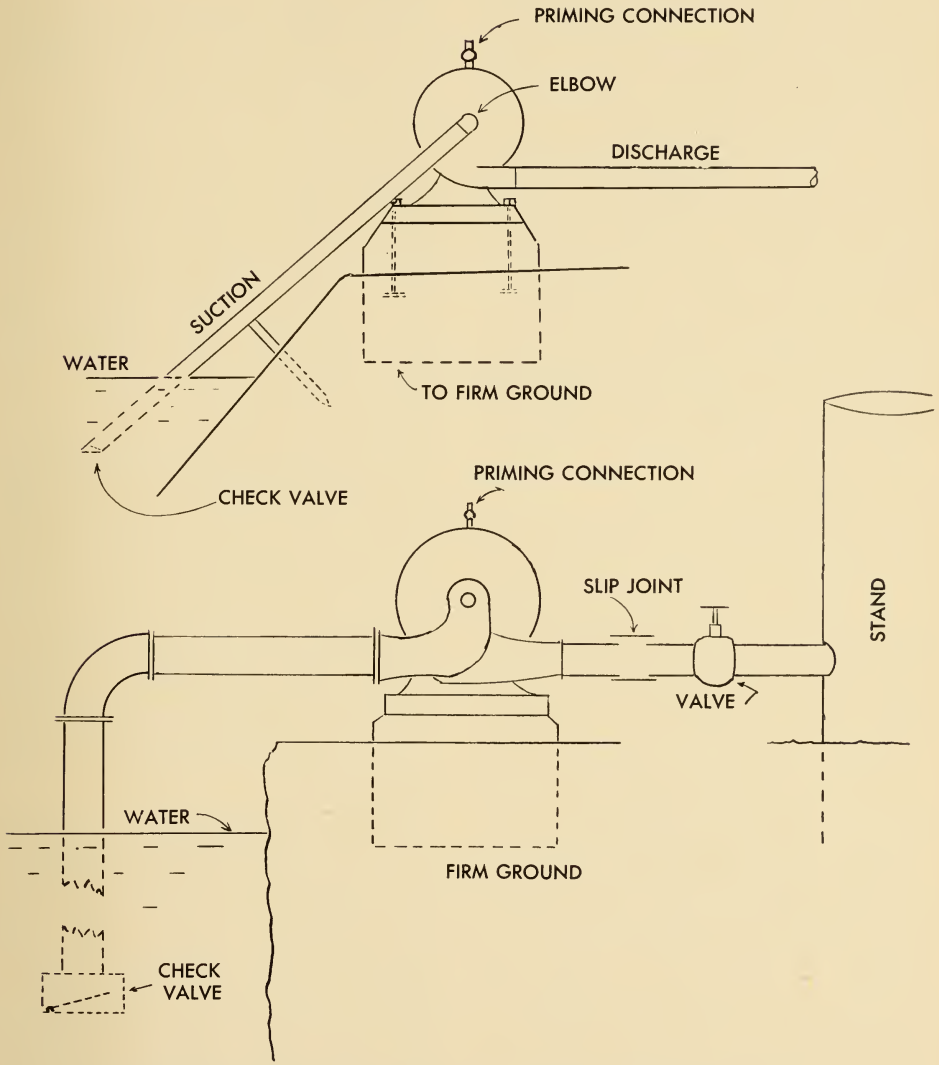
There are two reasons for simple piping arrangements: (1) simple piping is generally most economical; and (2) it usually includes straight runs and smooth bends, if any are present, both of which give best results when water is to be transmitted.

It seems, then, that the foundation is located to suit the piping, which is very often the case—with the piping in the form of plans and not coupled up at this stage. A good heavy concrete casting (See drawing) will be firm and lasting enough if sunk well into the ground in stable material. The hold-down or anchor bolts are cast in the foundation when it is poured. The top of the concrete casting need not be exactly level, but when the pump base is set over it the base should be leveled carefully, using metal shims to hold it at an exactly level position when it is bolted down, or to permit grouting around the whole base with a rich cement mortar. (This will lock the shims in place and provide an exactly fitting solid form for the pump base to rest on.) Under this procedure, the hold-down bolts from con-

crete foundation to pump base cannot distort the latter and throw pump and motor out of line.

The drawing shows such an installation and other details of the centrifugal-pump installation. It is to avoid any disturbance of the factory alignment of pump and motor that a good foundation is required, both at the time of installation and throughout the life of the pump. A solid, lasting foundation tends to eliminate vibration. This is mechanically beneficial to the unit, and unpleasant noises are reduced or eliminated from the sound-transmitting system afforded by the piping.

The piping can now be attached to suction and discharge of the pump, once the pump base is securely set on the grout, or bolted securely on the shim plates. In planning this piping the construction of the pump had to be considered so that suction and discharge ports could be oriented correctly when the pump-unit base was installed on the foundation. Below



Foundations and piping for two types of horizontal centrifugal pumps.

you see the pump end of a single-suction and a split-case double-suction horizontal centrifugal, with suction and discharge piping connected.

Suction piping. Starting with the suction pipe, one of the chief considerations in the general layout was to provide a satisfactory suction-pipe arrangement. This means that the pump was put near enough to the source so that the suction line could be as straight and short as possible. In both the pumps illustrated, an elbow or 90-degree turn is shown. This is standard practice, but by heating and bending the pipe in a smooth curve a longer-radius bend can be made, which would have less friction loss.

So-called long-sweep fittings are available at some pipe yards. These screw or flange together in assembly and are better on the suction line than the standard elbows. Straight, short suction lines provide as low suction lifts as possible. This is of great importance in centrifugal-pump installations. They will lift water 20 feet or more on the suction side, but they will operate much better if the gross suction lift can be kept below 10 feet.

On the bottom of each of the suction pipes illustrated on page 45 is a check valve. This is a flap valve that opens when water is drawn through it to the pump but closes when the tendency for flow reverses. This feature is made part of these installations because, by holding the water in the system, check valves, or foot valves, keep the pumps full of water (primed), and it is possible to start them at any time so long as they stay full of water. In the sketch a priming connection is shown at the top of both pump cases.

With a foot valve installed, it is necessary only to fill the suction line and pump with water, pouring it through the priming connection to prime the pump. The discharge line has to be closed off or else rise above the pump case in order to accomplish the result. Where no foot valve is provided the priming connection can be used as a suction inlet from a vacuum pump of some sort, if the discharge pipe has a shut-off valve handy to the pump.

In fact, that valve becomes necessary if no foot valve is installed. Assuming the discharge valve is closed and a common hand-pitcher pump suction is threaded into the priming connection, the pump case and suction can be partially evacuated of air. Water from the source will then follow the reduced internal pressure area to the pitcher pump, priming the centrifugal so that its motor may be started.

Leaks. If there are leaks in the pump suction or discharge, priming the pump by pumping out the air may be difficult or impossible, depending upon how great the leaks may be. Leaks at piping joints, if present, have to be taken up by tightening them further, or by applying heavy paint, grease, or other mastic material along the leaking joints. Leaks in the pump occur at the stuffing boxes, or seals, on the drive shaft. These stuffing boxes have a seal called packing inside them. The packing is held tightly by a sleeve and hold-down bolts.

With a new smooth pump shaft and resilient packing, the hold-down bolts or nuts need be only hand-tight to assure good air seals at these points. When it becomes necessary to force the packing down with greater pressure, there is danger of producing excessive friction on the pump shaft, which will cause scoring of the metal. When scoring begins a vicious circle has started, because it has become more difficult to hold a seal with the packing, more pressure is needed, and more scoring results.

Discharge piping. The discharge pipe from a centrifugal pump is sometimes smaller than the suction. This is so because the manufacturers wish to keep the suction lifts as small as possible, and a larger suction pipe promotes lower suction lifts. The suction pipe should be no smaller than the pump provides at the suction-inlet connection. The discharge pipe from the pump has less effect on its

operation than has the suction pipe on excessive suction lift. Therefore long-sweep bends are not fundamental. Discharge valves are needed if control of the discharge rate or other reasons dictate.

The slip-joint coupling shown on the split-case centrifugal discharge is mandatory when the steel discharge pipe is terminated in a concrete standpipe or similar relatively weak structure, as shown. Without the slip joint the temperature changes natural in the pipe will cause expansion stresses at the stand that may fracture it, causing leaks. The only other solution for such installations—a difficult one—is to make some sort of stuffing box in the stand within which the steel pipe can move without creating leakage.

One can observe from the drawing on page 45 that the split-case double-suction pump lines up suction and discharge nicely, but that the single-suction pump has the suction offset from the discharge. As we saw before, this latter type of pump may have the pump case rotated around the impeller by unbolting the case from the back plate bolts and rebolting it in the position that puts the discharge in satisfactory alignment. Sometimes in so doing the setting, or foundation, has to be switched around. Thus, in planning the arrangements of piping and foundations this fact must be given consideration. Occasionally it is possible to set these pumps so that the suction looks toward the supply and a curved pipe can be run from suction connection to source, providing a smooth, free-flowing entrance for the water. Curved pipes cannot be threaded into a pump directly but require a union or flanges at the pump.

Deep-well turbines. These, as we have seen, are relatively much more bulky than centrifugal pumps with their long column pipes, drive shafts, and so forth. The installation is, therefore, a special job requiring heavy lifting equipment, large clamps, heavy pipe tongs, etc. For this reason, the seller usually supplies outside help to make the installation.

Several precautions are taken in such installations. All joints are threaded, and great care is taken that all threads are wiped clean from dirt of any sort before they are lubricated with red lead or its equivalent and turned down in place. Equal care is taken that threads start accurately matched so that no cross threading is permitted to injure the starting turns. Every joint is set up tight, and the final adjustments of tubing and column length are made at the pump head where they terminate.

The pump head has suspended on it all the weight of everything below. Often the top of the casing is used as the supporting surface for the pump head. This is generally satisfactory if the casing is cut smoothly and at right angles to the casing length. Any tendency for the pump head to tip up or hold up from the squarely cut casing should be corrected by shimming so that no bending strain is permitted. If the casing itself has an angle off the vertical, this means the pump head will and should stand at this same angle, because in so standing it indicates a straight line from head through shaft—at least for some distance.

If heavy steel beams are provided for the pump head—or better yet a concrete foundation—similar precautions to those taken with the casing should be followed. The pump head should line up with the casing. These independent foundations are safer than use of the well casing as a support for the pump head. Heavy wooden timbers are sometimes used, but the tendency is to install and forget them, and time causes deterioration or shifting, creating misalignments that go unnoticed and unattended until mechanical failure results. They are not recommended.

Turbine discharge pipes. The discharge pipe from a turbine might duplicate that for the horizontal split-case centrifugal on page 45. The valve is optional. The slip joint is important when the discharge enters a concrete standpipe and, if that is the case, it might be very

desirable to supply a discharge flap valve on the end of the discharge pipe inside the stand.

Where a deep-well turbine discharges into a long line below the water surface, a large amount of water could run back down through the pump unless a flap valve stopped it at the time the pump was shut off. Water running down the pump column tends to turn the pump backwards, and this rotation can unscrew the drive shaft. The result of unscrewing the drive shaft is generally mechanically disastrous. Severe stresses result that may rupture the pump head, oil tubing, or column pipe—sometimes all three.

For this reason, a flap valve at the stand is necessary where back flow is possible. On a long, closed steel line, a check valve near the pump is quite important. A few pumps have nonreversing features built into the unit, but a check or flap valve provides a worthwhile safety factor for these units too.

Deep-well pump adjustments. The pump, after installation, must be "adjusted" so that the runners turn freely without touching either top or bottom of the bowls. This adjustment is of necessity accessible to the operator at the ground surface. It comprises a threaded section of the top end of the drive shaft fitted with two hex nuts supported by a heavy-duty ball bearing with a thrust plate.

This ball bearing, mounted firmly in the pump base or on the motor head, as the case may be, supports the complete weight of the drive shaft and impellers when the bottom nut is screwed down the shaft far enough to put the whole shafting assembly in tension down to the bottom bowl. If this bottom nut is screwed further down the drive shaft, the shaft is raised through the bearing until the runners touch the top of the bowl cases. By releasing the nut, the runners drop with the shaft until they seat on the bottom of the bowl case. The "adjustment" is at some point between top and bottom positions for the impellers in the bowls.

In reality the runners as a rule perform best when they turn freely as close as possible to the bowl seats, or when they are as low as they can be made to turn freely when in operation. This is often determined by trial-and-error procedures in the field. It is best to have the "adjustment" of the runners made by an expert and not attempt it on one's own initiative. A check of the adjustment is afforded by the watt-hour meter which will record a greater use of electrical power if the runners are rubbing than when they are free.

These adjustments are especially important for runners having the lower side open, or unshrouded. With this type, wear on the exposed vanes can be compensated for by lowering the runners a few thousandths of an inch, and efficiencies often respond by an increase of 10 to 20 per cent overall. Closed impellers having two shrouds do not benefit particularly by the critical shaft adjustments just described. Submersible pumps obviously cannot be "adjusted" while in the well.

Other types of pump drive

Direct motor-driven electric pumps are most common, but other forms of drive and sources of power are available. Belts are one of the most common forms of transmission used for driving pumps. The horizontal centrifugal generally has a horizontal pulley. Therefore a straight running belt (no twist) needs to be used from any horizontal shaft-power source (including motors, fuel engines, and power take-offs from tractors). Deep-well turbines are vertical, so that the pulley mounted on the drive shaft stands vertically. As a result, horizontal-shaft power units have to be connected with these pulleys by putting a 90-degree twist in the belt.

Sometimes enclosed geared boxes are mounted at the top of the pump head with a pulley shaft parallel with the ground. These permit coupling internal-combustion engines or other power units to them directly through flexible or universal

couplings. It is possible to drive such geared heads with a straight belt from a horizontal-shaft power unit. The coupling between the fuel engines and geared heads should be resilient to take up the shock of the power strokes of this type of drive unit.

Certain advantages are inherent to belt-drive units or to fuel-engine driven units. The revolutions per minute of the pump can be changed relatively simply by changing pulley sizes in the first type, and by simple throttle adjustments for the second. The problem of overloading discussed under equation 3, page 10, is always a possibility where increased speed is given the pump. Thus it is well to watch the driving unit carefully for a while after such a change has been made. Overheating is the sign of such overloading in most power units.

Tables 1, 2, and 3 give the horsepower capacities of flat and vee belts. The velocity of a belt in feet per minute is computed as follows: Feet per minute = rpm of drive pulley × diameter of drive pulley in feet × 3.1416. For example, a 6-inch-diameter pulley is turning 1,740 rpm. How fast is it driving the belt?

Solution: $1{,}740 \times \frac{1}{2} \times 3.1416 = 2{,}733$ feet per minute

The second table, for quarter-turn or crossed belts, shows the space necessary between drive and driven pulleys. It is essential to provide this spacing for crossed belts so that they do not rub and so that the transition from the horizontal to the vertical pulleys can be made gradually. Under best conditions, the spacing of two horizontal pulleys should be about the same as that for the crossed belts. The reason for the longer spacing shown for engine drive is the shock effect contributed by this source. Vee belts can be run straight or crossed.

Table 3 shows the capacities of five sizes of vee belts—through E—and the belt speeds possible. The vee belt clings to the pulley and therefore does not have to be set up so tightly as the flat belt, which

Table 1. Horsepower Transmitted by Belting

Belt speed, feet per minute	Double-Ply Leather Belts		Rubber belting with varying ply values							
	Horsepower per inch of width									
	Single-ply	Double-ply	3-ply	4-ply	5-ply	6-ply	7-ply	8-ply	9-ply	10-ply
1,000	1.4	2.0	1.25	1.67	2.08	2.50	2.92	3.33	3.75	4.16
1,500	2.1	3.0	1.88	2.50	3.12	3.75	4.38	5.0	5.64	6.24
2,000	2.8	4.0	2.50	3.33	4.16	5.0	5.85	6.66	7.50	8.33
2,500	3.5	5.0	3.12	4.16	5.20	6.25	7.30	8.33	9.40	10.40
3,000	4.2	6.0	3.75	5.00	6.25	7.50	8.75	10.00	11.30	12.45
3,500	4.9	6.9	4.35	5.80	7.20	8.75	10.20	11.65	13.15	14.55
4,000	5.6	7.9	5.00	6.65	4.30	10.0	11.70	13.30	15.00	16.65
4,500	6.3	8.9	5.60	7.50	9.35	11.25	13.15	15.00	17.90	18.70

Table 2. Belt Sizes for Quarter-turn Drives

Horsepower transmitted	Leather belts		Rubber belts		Minimum center distance	
	Width	Ply	Width	Ply	Motor drive	Engine drive
	inches	number	inches	number	feet	feet
10	4	1	5	3	10	15
15	5	2	5	4	12	18
20	6	2	6	4	14	22
30	8	2	8	5	16	25
50	10	2	10	5	18	30
75	12	2	12	5	20	34
125	16	2	16	6	24	37

will slip unless reasonably tight. This slippage causes a waste of power. Some slip is almost inherent with flat-belt drives.

Fuel engines including automobile engines are used for pumping. Only the heaviest-duty engines of this type can stand continuous full-load operation. Automobile engines cannot. To supply 40 horsepower from a car engine one should have one capable of giving 80 horsepower intermittently (this is the way they are rated for intermittent service).

Electric motor sizes and efficiencies: Motors can be bought in only certain stock sizes, as listed in table 4. The approximate efficiencies at full load or rated load are also detailed in the table. The motor rating is the horsepower it will

Table 3. Horsepower Transmitted by Horizontal V-belt Drives

Belt speed, feet per minute	Horsepower per belt				
	A	B	C	D	E
1,000	0.9	1.2	3.0	5.5	7.5
2,000	1.7	2.3	5.5	10.0	14.0
3,000	2.4	3.2	7.5	14.5	19.5
4,000	2.8	4.2	9.0	17.5	23.5

Table 4. Approximate Efficiencies of Induction Motors at Rated Load

Motor size, horsepower	Efficiency, per cent	Motor size, horsepower	Efficiency, per cent
5	86	30	90
7.5	88	40	91
10	89	50	91
15	89	60	91
20	89	75	91
25	90	100	91

deliver on the drive shaft. Thus if the motor is delivering 45 horsepower at 90 per cent efficiency it is actually drawing 45/.90 = 50 h.p.

Most motors can be put in service to deliver up to 10 per cent overload, which means to deliver 11 h.p. from a 10-h.p. motor, 22 from a 20-h.p. motor, and so on. For this reason, from the table, a 40-h.p. motor could be loaded to 44 h.p.; but if 45 or 46 h.p. were the load, then a 50-h.p. motor should be used. Overloading causes heating, and heat ruins insulation in most motors, shortening their lives. For this reason it is not recommended that any overload be put on them.

CARE AND MAINTENANCE OF THE PUMPING PLANT

Since a pumping plant is relatively expensive, the effort to keep it in best condition is justified. Care and maintenance are immediately in order once a satisfactory initial installation has been completed. Housing comes first to eliminate dust and dirt and to keep out precipitation.

The size of the pump house is determined by the equipment, it being necessary only to provide working space all around it. Three feet clearance is generally enough. When the pump is running, ventilation is required to take off the heat developed by the power unit. Therefore a door and windows that can be opened are generally installed. Sometimes louvres are put in the sides of the pump house to permit continuous ventilation, with cross draft if possible.

In cramped quarters, servicing any equipment is difficult. When the operator has to wedge himself into too cramped an area to perform a regular servicing task, it may seem easier just to forget the job. Then lubrication or some other attention is omitted until injury results to the installation.

All-weather motors are available that require no housing. However, adequate housing would appear to offer some benefit to such units by shading and protection from dirt. Both pump and motor bearings need oiling, and the oil containers provided by the manufacturers should be filled regularly so that the supply never runs out.

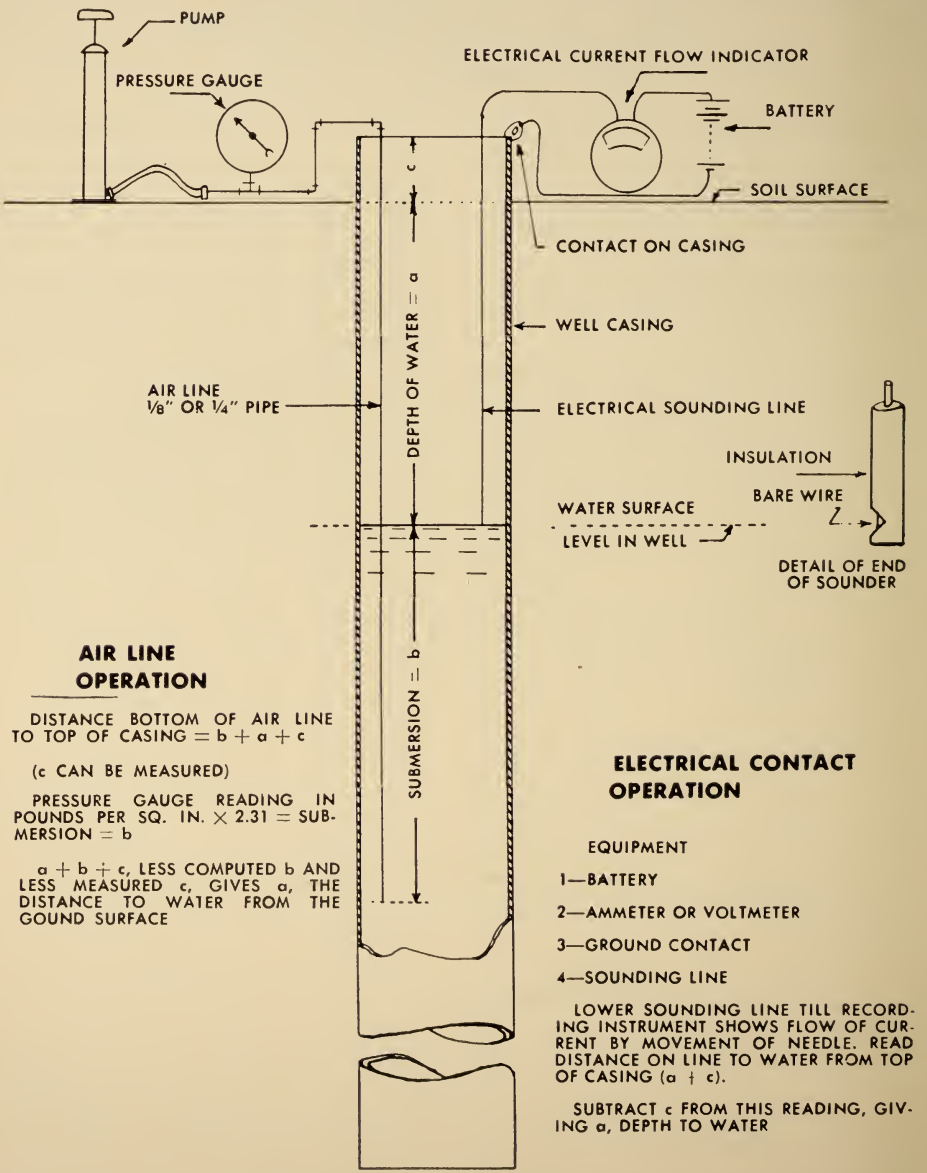
As the pumping plant ages, check the performance occasionally to be sure overall efficiencies have not dropped so low that power costs have become excessive. In addition, a certain amount of clean-up is desirable on the equipment, both for appearances and (more important) to eliminate the possibility of dirt working into the moving parts.

Keep stuffing boxes tight but at the same time draw down the take-up nuts as lightly as possible. After several thousand hours' operation, repacking of the stuffing boxes may be necessary. Sometimes the addition of a new ring of packing is all that is needed. It is safer, if the old packing has become hard or grit has been imbedded in it, to remove the old and put in new. Packing has a lubricating material in it and is obtainable through the pump supply houses in the locality.

Most power companies willingly check the performance of electric-driven pumps on their system, and it is generally possible for any customer to obtain this service. These operators with power-company test service, and others who are not so supplied, may wish to check well-water levels occasionally to see what the underground water supply situation is. The drawing on page 52 illustrates two methods that may be used in "sounding" these wells to determine static-water and pumping-water depths. The two methods are, respectively, an air line and a measuring tape or line.

In the drawing the measuring line is an

insulated wire which completes a circuit with a battery and a meter of some sort in series with it. One can substitute a magneto for the battery in the wire sounder and have a bell ring when contact with the water is made. A tape line or a string with a small disc or can on the end may sometimes be used to sound a well if there is enough room for them to pass down between the column pipe and the casing. The tape line is run down so that the end is well immersed in the water. Then the net dry length is noted to get the depth to water. The can or disc can be lowered to the water surface, and it is generally possible to feel them make contact. If they submerge, the net dry length can be measured as with the tape line.



Devices for measuring depth to water.

Tapes fail for this purpose when water is leaking into the casing above the water surface in the well. The falling water wets the tape and obscures the wetted mark due to submersion. It is to preclude effective continuous contacts with falling water on the electrical sounding wire that the terminal is guarded as suggested here.

Actually, apparent contacts with the water surface will result from introduction of the terminal into a continuous falling stream, giving surface indications that otherwise would mean the water surface had been reached. However, when the line is moved up or down, the indicator will show erratic readings in these falling streams. The readings suddenly become steady when the true water surface is reached.

The air line is, as shown in the sketch, a small pipe run down some distance below the lowest water level in the well. It has an open end at the bottom, and, if a pump is connected to the closed upper end, it can produce sufficient pressure in the pipe to force the water out. Any extra air put into the pipe above that pressure will pass out the bottom of the pipe. The pressure read in pounds per square inch (from formula 4) can be converted to feet of submergence of the pipe. All one needs to know, then, is the total length of the pipe to calculate the depth to water, thus: Total length of air line (soil surface to bottom of pipe generally) minus submersion = depth to water.

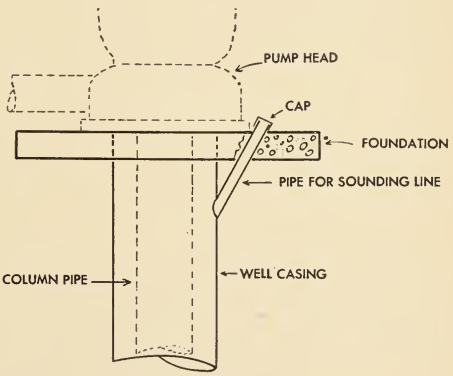
Some pressure gauges can be set to read directly the depth to water in feet, by providing reversed scales that can be set to read the length of the air line when the needle is at zero pressure. Air lines and tapes cannot be used in some wells because there is no clearance for them, because the well is too crooked, or because the pump head provides no inlet for this purpose. Under these conditions, efficiency tests of the pumping unit are impossible either for the individual or for anyone else.

On new, deep-well pumping-plant installations the owner can provide a means for measuring the depth to water by providing a pipe, as shown in the drawing (below), that leads from the ground or foundation elevation to and through the side of the casing. By making the edges of the pipe ends smooth so that the measuring line will not be scratched or injured, and by providing a screw cap for the top of the pipe, entrance to the well and the water levels is always possible.

None of these difficulties beset the normal horizontal-centrifugal setting because accurate vacuum- and discharge-pressure gauges can easily be connected to them and total lifts can readily be obtained.

One of the indications that water levels are getting too low for the pump is the discharge of air, or intermittent discharge from the pump. When this stage is reached with a deep-well turbine, the pump must be lowered 20 to 40 feet to put it down in the supply deep enough to provide safe continuous operation. This step may involve reworking the bowls at the factory and perhaps the installation of a new motor and drive shafting.

Failing discharge from a horizontal centrifugal with air in the discharged water may be the result of leakage of air through the stuffing box. This is more apt to occur when the suction lifts are excessive.



Small pipe with capped cover provides special entrance for sounding line in wells otherwise impossible to sound.

If water appears in the container on a deep-well turbine-drive shaft oiler, there is reason to believe the oil tubing has corroded through or has failed in some way. Bearings will not last long in a pump so affected.

Settlement in foundations or piping as time passes can throw excessive strain upon pump castings, causing distortion or failure by cracking off parts of the affected structure. Repairs to obvious changes in alignment of parts resulting from these natural movements may save many dollars.

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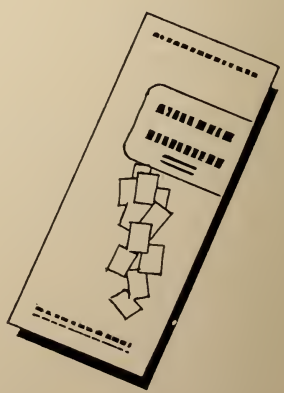

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